PRODUCT ENGINEERING

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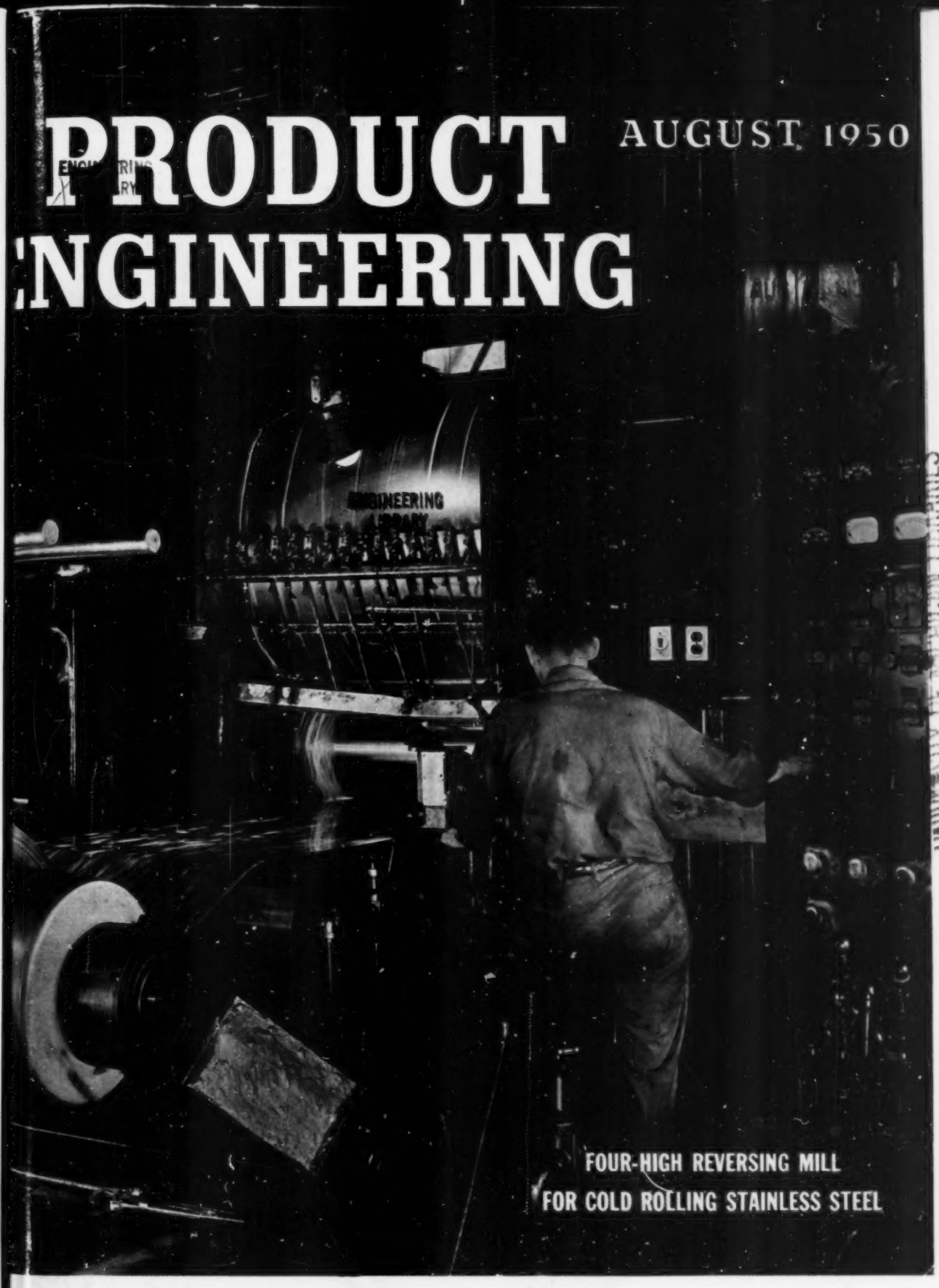

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IN THIS ISSUE — Aluminum Forging Design ✦ Electrical Bi-metals ✦ Surface Finish Control ✦ Acceleration of Hydraulic Motors ✦ Circular Arc Cams

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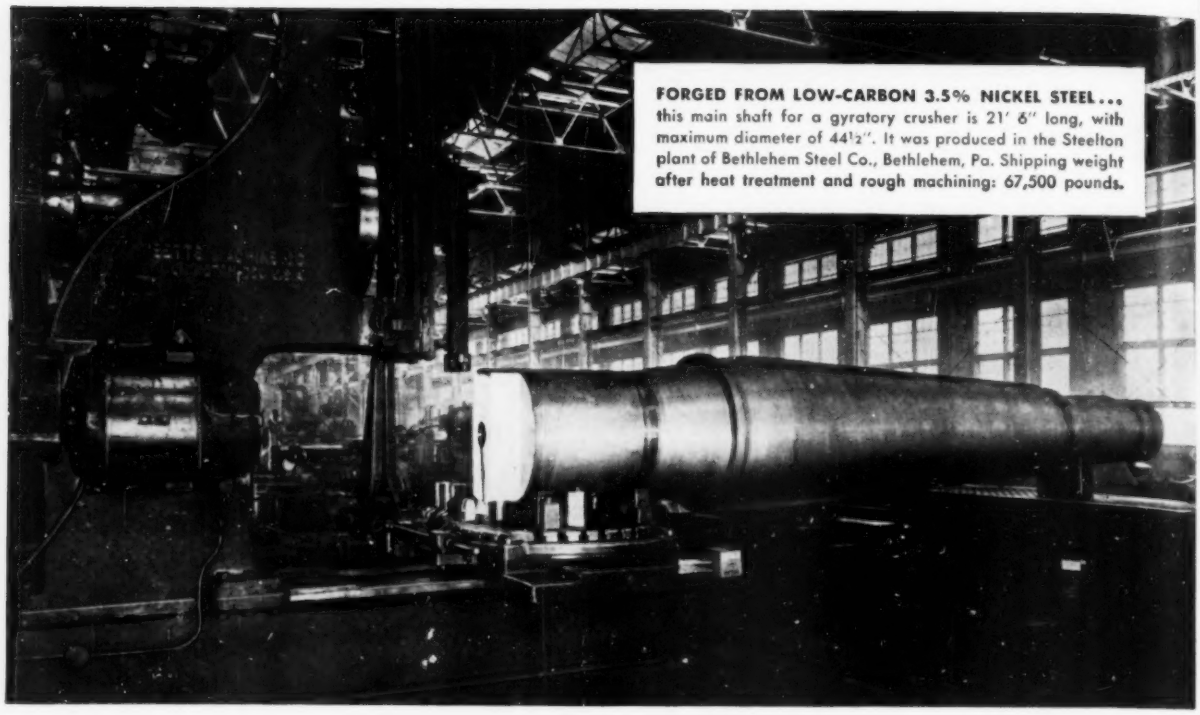

FORGED FROM LOW-CARBON 3.5% NICKEL STEEL... this main shaft for a gyratory crusher is 21' 6" long, with maximum diameter of 44½". It was produced in the Steelton plant of Bethlehem Steel Co., Bethlehem, Pa. Shipping weight after heat treatment and rough machining: 67,500 pounds.

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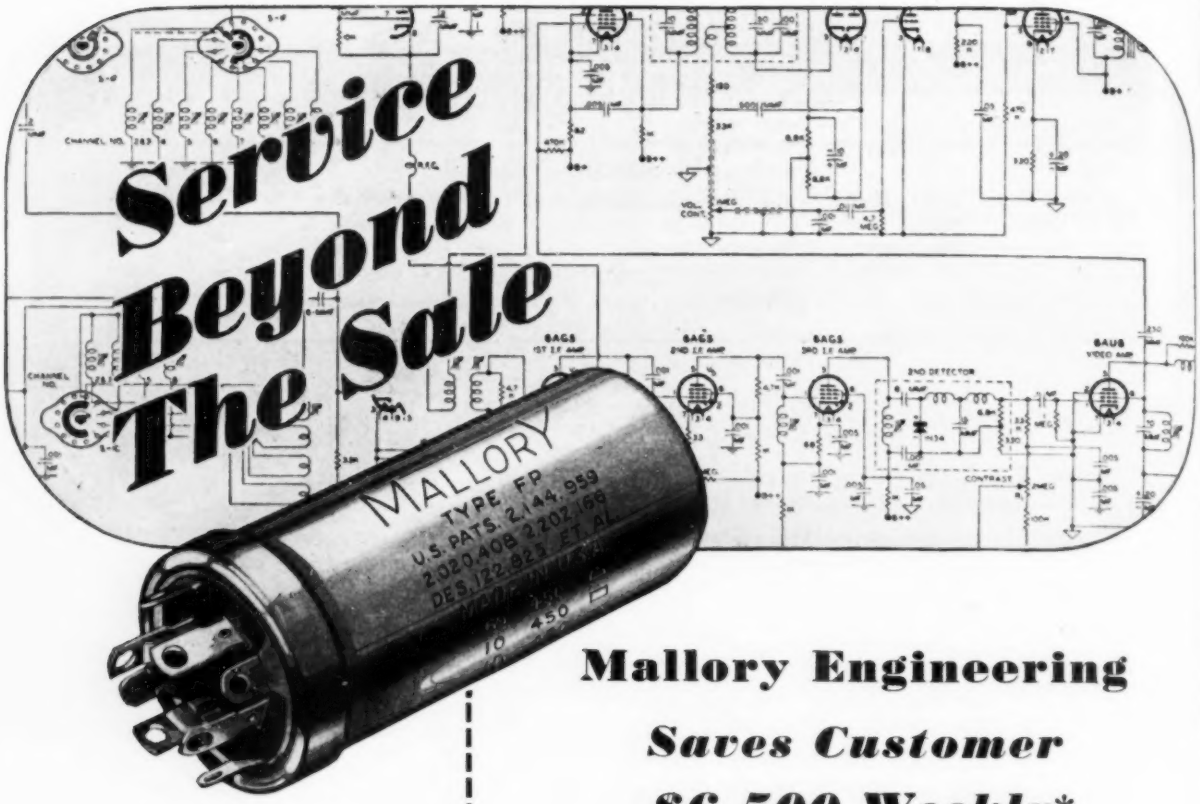

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The Lorelei Song for Lower Prices

BECAUSE LOWER PRICES TEND to increase the market and make easier the selling of product, there is often a strong urge, particularly in the sales department, in the direction of lower prices. In addition, there is also the desire for a wider margin of profit, and hence the generally proposed method for achieving the end result is not to raise prices, but to lower manufacturing costs.

There are many ways of cutting manufacturing costs without reducing the quality of the product. Sometimes it is even possible to improve a product and at the same time reduce the manufacturing costs. But eventually a limit is reached and it becomes impossible, at a given time, to further reduce production costs without seriously affecting the quality of the product.

Purchasing and sales, as well as other non-engineering personnel of a manufacturing company, are often extremely vehement in their "suggestions" for cutting the cost of a product at the expense of quality. When it is pointed out that their suggestion will result in an inferior quality they often belittle the value of quality and brush aside the objections. Almost every time that a product of high quality and distinction has been cheapened it has later lost its leadership.

The market for a quality product exists primarily because of its quality. People buy high class, top-ranking goods primarily because of qualities of some kind or other—appearance distinction, serviceability, durability, convenience, or other feature. Ironically, only too frequently the people who sell the product often fail to recognize the qaulities which give it sales acceptance.

This last possibly explains the all too prevalent attitude on the part of those who urge a lowering of quality standards with the argument, "what difference does it make? The customer won't know it." Such a stand is not only an insult to the intelligence of the customer but also underrates competitors. Invariably the first to learn about lowered qualities are the competitors who immediately tell the customers and prospects.

When what was a quality product appears shabbily clothed or no longer able to perform with superior satisfaction the functions for which it was designed, or no longer carries with it a pride of ownership, because it has lost its earmarks of distinction, the fact will very quickly become known generally. Competition will shout it from the housetops; customers will grumble or bellow about it. If the product could speak it might quote Shakespeare in saying ". . . he that filches from me my good name robs me of that which not enriches him, and makes me poor indeed."

G. F. Modenhott

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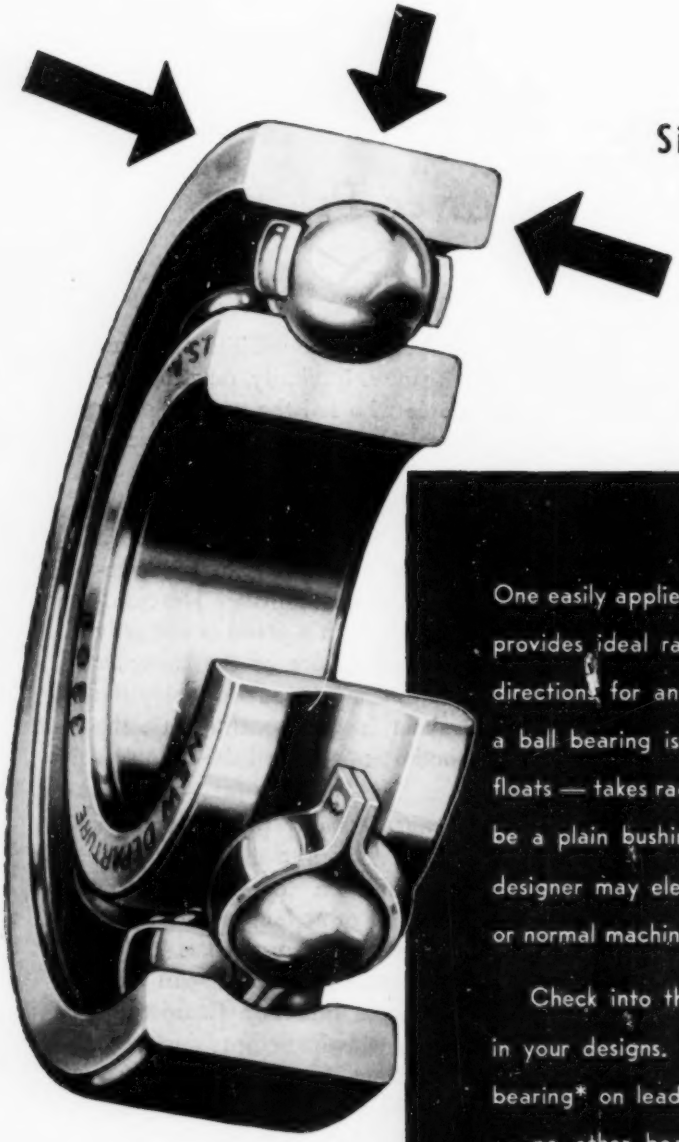

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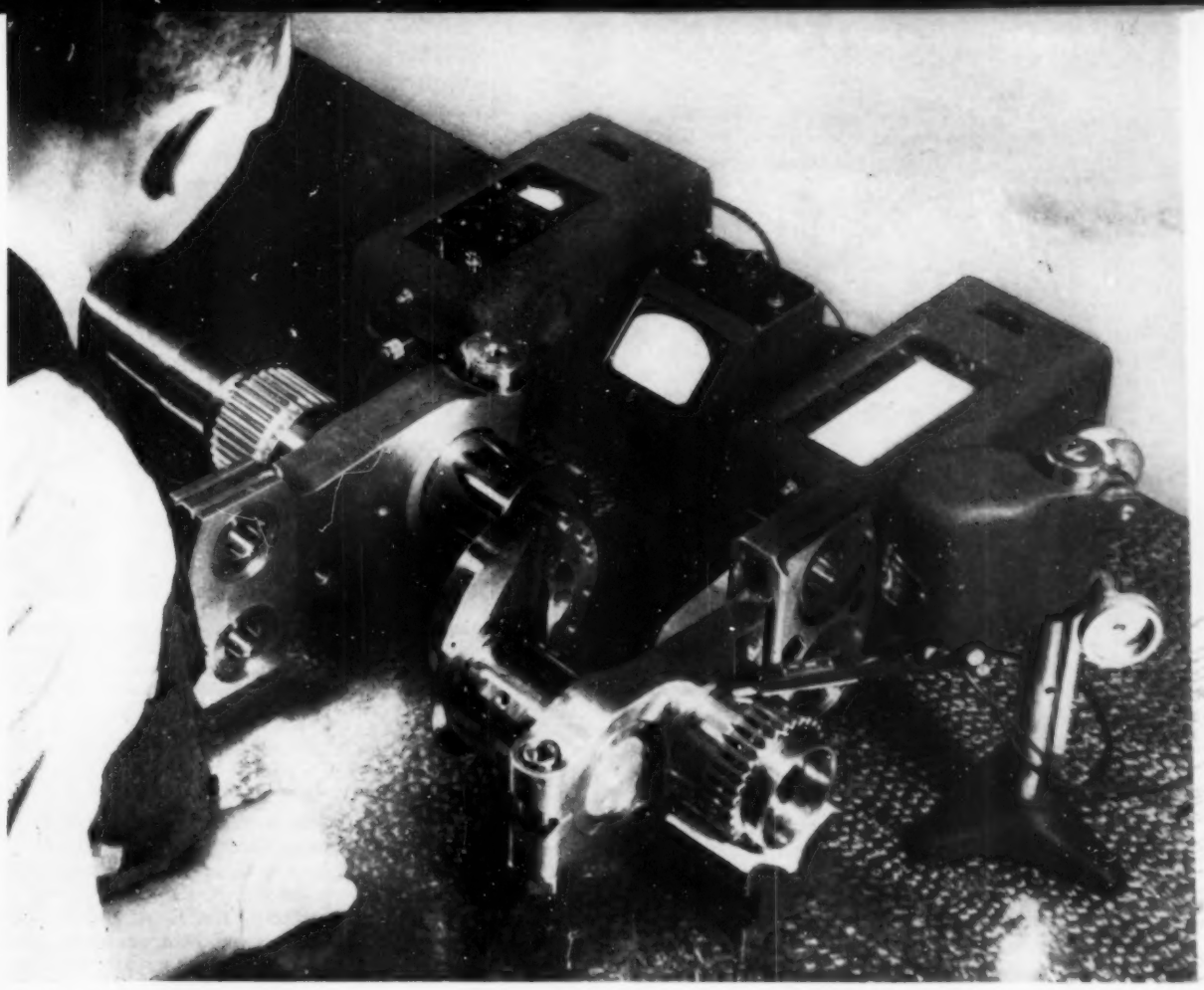

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Surface Finish Control

A realistic approach to the old problem of surface finish specification, including a discussion of new SAE standards and of the various effects below the surface of metals caused by different finishing operations.

DR. C. R. LEWIS, CHRYSLER CORPORATION

To obtain maximum benefit from Dr. Lewis' article, the reader should turn to the surface finish standards presented as Reference Book Sheets, pages 163 to 167. Although this may seem inconvenient, it was felt that engineers interested in surface finish would want to clip the new standards for future use. They are thus presented separately.

THERE IS NO SIMPLE FORMULA that will allow a designer to choose an optimum surface finish for a given part. The only safe way of specifying surfaces to obtain minimum wear and optimum performance is to base such specification on surface conditions previously found satisfactory. However, the designer should have some working knowledge of the surface finishes that are obtainable from common production operations and of the results to be expected from the use of such finishes. Furthermore, he should know how to specify the type of finish required so that the results he expects are obtained by the shop.

Surface finish measurement and control, as practiced at the present time, is confined to the departure of an ac-

Fig. 2—Taper sections of surfaces produced by common finishing operations: (A) Milled surface, magnification horizontal 700X, vertical 17500X; (B) fine turned surface, magnification horizontal 550X, vertical 13750X; (C) ground, magnification horizontal 400X, vertical 10000X. Vertical exaggeration, because of the tapered surface from which the photographs were taken, provides a definite comparison between the various finishes. Fig. 1 shows sectioning technique.

tual surface from the geometry of an idealized surface. To systematize such measurements, a new standard has been developed by the Society of Automotive Engineers, and will be published as the surface finish section of the new SAE Automotive Drafting Standards.

To study and illustrate surface finishes, the taper sectioning method proposed by Nelson has come into general use. As shown in Fig. 1, the surface

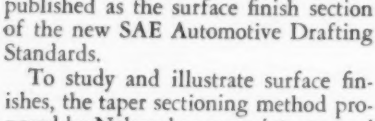

Fig. 1—Technique of taper sectioning for examination of surface irregularities allows vertical magnifications from 10 to 25 times as great as the horizontal magnifications.

to be viewed is cut at a very small angle to the original surface of the specimen. The taper can give magnifications as high as 25 to 1. In viewing pictures of taper sections it must be remembered that the photographs are distorted, with the vertical magnifications 10 to 25 times as great as the horizontal magnifications. Since the heights of the surface irregularities are exaggerated in relation to their widths, the average slope found on an actual surface is always much smaller than those shown on taper sections representations.

Photographs of taper sections of surfaces produced by common finishing operations are shown in Fig. 2. Machined surfaces vary so widely that these illustrations should not be taken as typical of any type of finish, but only as illustrations of particular surface finishes. The distortion in these pictures must not be forgotten. For example, the surface shown in Fig. 3, which was made by a special method as a surface roughness standard, has an included angle in the grooves of 150 degrees instead of the apparent angle of about 15 degrees.

Effect of Finish on Metallic Structure

Surface finishing operations are suspected to have an important effect on the structure of metallic materials at or immediately below the finished surface. It has been contended that operations such as polishing or the running-in of a surface in actual operation produce a layer of non-crystalline metal. This is still controversial, and existence of such a layer has been neither established nor disproved.

When rough surfaces are produced, grain distortion and fragmentation occur. This distortion is clearly visible in Fig. 2B. Even on a "smooth" surface, effects of finishing operations may extend to appreciable depths. When the polished surface of a metal is removed by etching, residual effects of previous finishing operations are often discovered. The well known police method of recovering obliterated serial numbers on guns depends on this effect.

To show the effects in depth of surface working, a number of investigations under controlled conditions have been made, of which three typical results will be described. Using X-ray methods, Thomassen and McCutcheon investigated the depth of cold working in 70-30 brass (surface roughness presumably of the order of 100-200 microinches) when milled with a feed of 0.007 in. per revolution. X-ray patterns showed cold working on these samples to a depth of 0.005 inches.

Using an electron diffraction method Wulff made a similar series of tests using initially smoother surfaces, and materials of such a type that both cold working and hot working could be detected. Commercial grinding operations, resulting in a surface finish of about 2.5 microinches average, showed hot working to a depth of about 0.0002 in. for wet grinding and

Table I—Effect of Surface Roughness on Corrosion Resistance

(stainless steel samples exposed to hot salt solution)

Sample Preparation	Roughness (microinches average)	Weight Loss 0 to 24 hr	Weight Loss 48 to 72 hr
		mg/sq cm	mg/sq cm
Rough Ground	30	1.4	0.64
Find Ground	2	0.78	0.41
Superfinish	1	0.37	0.32

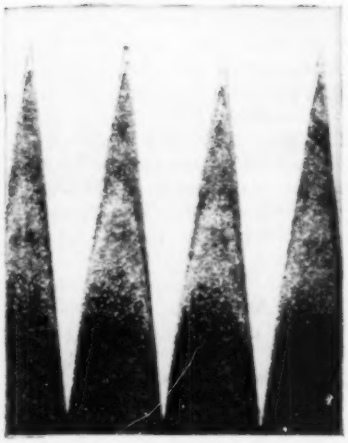

Fig. 3—Special standard surface (mag. horizontal 250X, vertical (250X) has an angle of 150 deg instead of 15 deg apparent angle.

nearly double this value for dry grinding. The temperatures at the bottom of the hot worked layers were about 400 F. Below this hot worked layer in both types of grinding was a cold worked layer extending to an additional depth of 0.0001 in. Smaller speeds and pressures affected less depth of metal. The combined hot and cold layers due to a honing operation reached a depth of 0.00008 in. and those due to metallurgical polishing about 0.00003 in. Since the method used for these studies required an initially smooth surface, no results are available for other machining methods.

These tests were made on 18-8 stainless steel, to measure effects of hot and cold working by resulting phase transformations. Further tests were made using Hadfield steel and beta brass, with similar results. Results of a same order of magnitude can be expected for other common engineering materials.

This work shows that even mild finishing operations can have an effect on metals to a considerable depth below the actual surface. Where the metallic structure at a surface is important, as in applicatons requiring corrosion or fatigue resistance, a finishing method should be chosen which minimizes these effects. As part of this study of surface metallurgy, corrosion resistance of stainless steel samples exposed to a hot salt solution was measured by Wulff as a function of surface roughness. As shown in Table I, the smoother surfaces were more successful in preventing the initial incidence of corrosion, but once started, the rate of corrosion is not as strongly dependent on surface finish.

A somewhat more spectacular metallurgical effect of surface finishing operations has been reported by Sakmann, who studied ground samples which had undergone friction tests. Although most of the samples showed only profile changes, in several of the samples regions were discovered which had undergone phase transformation involving volume increases. Sections of material were thrust up above the general level of the surface. These eruptions consisted of extremely hard and abrasive-resistant material. These transformed regions caused high friction values and difficulties in lubricating the running surfaces. Since these eruptions appear only on samples prepared by severe grinding, their formation can only be attributed to the method of surface preparation.

Sufficient work has been done to show that surface finishing methods have an effect on the structure and properties of engineering materials to a significant depth below the nominal surface. These effects should be considered when surface properties of the material in a proposed application are important, such as wear resistance.

Performance of Machined Parts

Surface finish of a machined part has some effect on the friction, lubrication, wear, and fatigue endurance characteristics.

In investigations to determine the effect of surface roughness on the performance of bearing surfaces with adequate lubrication, it is convenient to group the operating variables of oil viscosity, Z, rotating speed, N, and pressure on the bearing, P, into a single composite variable

$$S = \frac{ZN}{P}$$

When plotted against the coefficient of friction this variable shows the performance of the test bearing.

To make such a plot general in application, the friction coefficient is multiplied by the clearance ratio D/C, and the variable S by $(D/C)^2$, where D is the diameter of the shaft and C is the difference between diameters of the shaft and bearing. When represented in this way, a typical friction test of a steel journal running in a babbitt bearing gives results of the type plotted in Fig. 4. The maximum performance of the bearing occurs at the minimum value of the curve, and the values of S for minimum coefficient of friction can be taken as a figure of merit for the bearing. Table II, which lists performance values obtained from test bearings having journals with different surface finishes, shows that performance is improved by making the journal surface smoother.

Tests of a flat slider running against the surface of a plane disc gave similar results. Fig. 5 shows a family of curves of friction coefficient vs operating variables for a series of surface finishes. Here, the clearance ratio no longer appears and the square root of the composite variable S is used, be-

Table II—Effect of Surface Finish on Bearing Performance
(steel journal running in babbitt bearing)

Shaft	Type of Finish	Surface Roughness microinches	Clearance Ratio D/C	Value of $(D/C)^2$ ZN/P at minimum friction coefficient
A	Turned	130	1,150	0.38
B	Ground	70	1,200	0.25
C	Ground	50	630	0.06
D	Ground	14	520	0.0017
E	Ground	13	540	0.0013
F	Ground	7	500	0.0012
H	Ground	4	480	0.0010
I	Superfinished	1	500	0.0008-0.0011

(from article by Burwell, Kay, Van Nymegen and Morgan — see References)

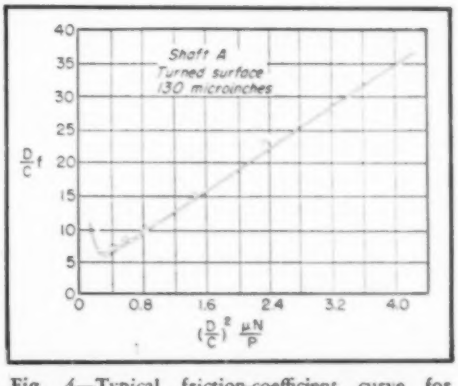

Fig. 4—Typical friction-coefficient curve for turned surface running against a babbit bearing.

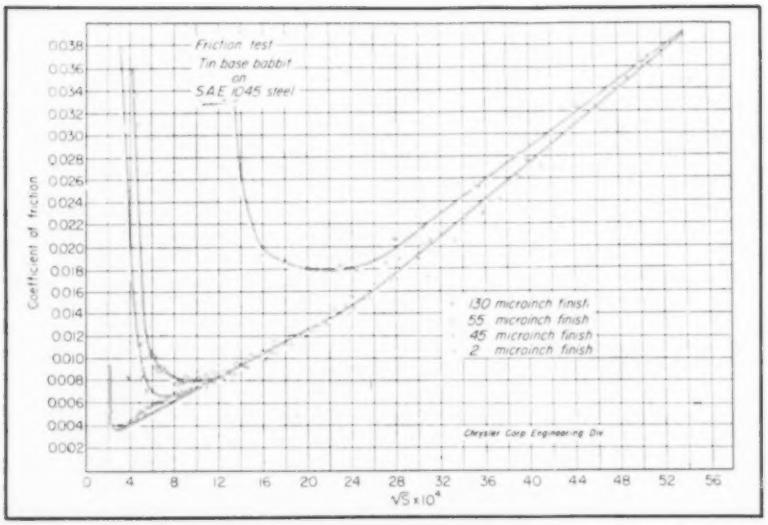

Fig. 5—Family of curves of friction-coefficient vs operating variables for a series of surface finishes as determined by various load conditions of an SAE 1045 steel journal.

cause the surfaces are plane rather than cylindrical. Table III gives the minimum values of $S^{\frac{1}{2}}$ as a function of surface finish, and also the corresponding loads carried by the bearing, assuming that speed and oil viscosity remain constant.

If lubrication of a bearing is restricted, the performance criterion usually considered is not friction but wear. Here again, tests show improved performance of smooth surfaces. In journal bearing tests of steel against babbitt, the removal of iron from the steel journal was measured. Fig. 6(A) shows that the initial wear increases as the surfaces are made rougher, but after approximately 20 hr of runnir , the rate of wear decreases to such a low value that no further distinction can be made. This test is primarily a measure of running-in wear, and not the normal wear of a bearing surface during extended operation. Fig. 6(B) shows running-in wear as a linear function of roughness.

Table III—Load for Maximum Efficiency

(SAE 30 oil at 170 F)

Surface Roughness microinches	$S^{1/2} \times 10^{-4}$ (min)	Load, psi
130	21.8	90
55	9.6	588
45	7.3	900
2	2.7	5,490

There has been some controversy concerning the effect of surface finish on the performance of such moving parts as bearings and gears. The difficulty arises from the word "performance". There is quite a difference, for instance, between performance during the "running in" period and performance after this period. This difference was demonstrated in tests of cylindrical cups which were rotated at different speeds under load, in a combined rubbing and rolling action that is typical of gear teeth. SAE 90 grade automotive gear lubricants were used. These tests were conducted at the National Bureau of Standards (McKee et al—see References). Two types of data obtained from these tests are worth noting.

To show the changes in surface roughness during test runs, starting with cups of the same roughness (about 28 microinches), different loads and various lubricants at 225 F were used. Some of the results are shown in Fig. 7(A). Where the rate of wear was relatively low (all lubricants but AS—this was determined previously), the roughness decreases during the run-in period and tended to level off between 10 and 20 microinches. As the rate of wear increased (lubricant AS—previously determined in time vs wear test), there was a trend for the roughness to increase.

Even more significant was the part of this study which showed the effect of the original surface roughness of the cups on the rate of wear. Surface roughness vs time curves, shown in the upper part of Fig. 7(B), indicate that the net result of operation was to bring the roughness of the cups fairly close together. Wear vs time curves in the lower part of this figure show that while the run-in wear tended to increase with increase in roughness, the final rate of wear was practically independent of original surface roughness, over range covered.

Where severe operating conditions are encountered, a protective coating is put on one of the surfaces to prevent damage during the initial running. Fig. 8 shows test results obtained on a coating of this type, using the same apparatus for obtaining the results in Fig. 5. Two curves are shown, one for uncoated cast iron surfaces and the second with one phosphate coat. The tests were started on the right side of the curve with very light loads, and the loads increased according to a definite schedule, thus decreasing the value of the variable S. The lubrication for the coated surface started to fail at a value of S characteristic of its roughness (compare Fig. 5). However, on continued running, the friction decreased to a value approaching that of the uncoated surface. Beyond the minimum of the curves the two lines approached each other, and the phosphate coating was practically worn off. A repeat of the test gave identical curves for the

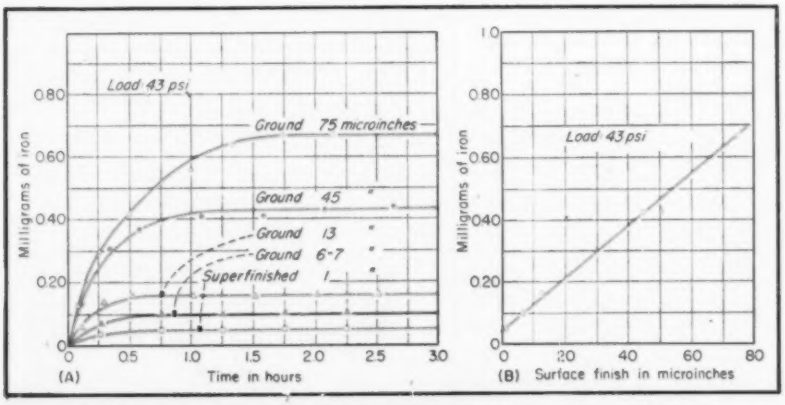

Fig. 6—Metal removal from steel journals (against babbitt) during running-in process: (A) Rate of metal removal; (B) total metal removed against surface roughness.

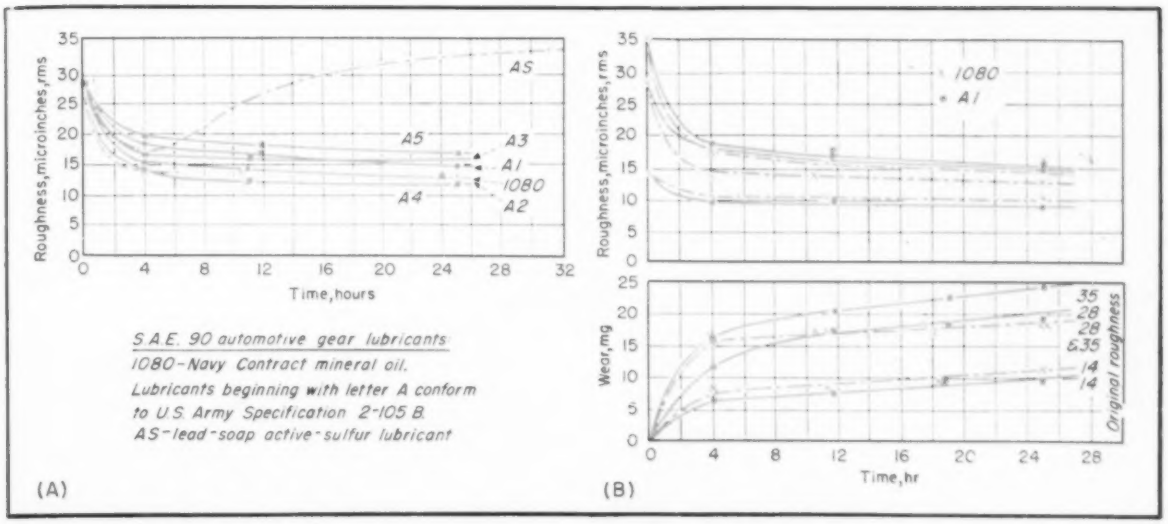

Fig. 7—Difference between running-in and later period by tests simulating automotive gear conditions (135 lb load and 225 F): (A) Surface roughness vs time; (B) surface roughness and wear as functions of time for lubricants, and surface roughness.

uncoated and run-in coated samples.

Thus, the presence of a "protective coating" does not increase the maximum performance of a bearing surface, but does allow rapid running-in, and protects the surface from damage during the running-in process.

In addition to influencing friction, wear, and lubrication, surface finish has an effect on fatigue endurance. Fig. 9 shows fatigue endurance test results for two samples of the same material having different finishes. The smooth finish has the higher limit.

Do You Need a Smooth Surface?

Variation of dimensional tolerances and machining errors contribute to a certain amount of distortion and misalignment when the part is put into a complete machine; in addition, thermal distortions usually appear when the machine is operated. If a bearing surface contacts its mating part only at one point or along one line, the small running-in wear found on smoother surfaces means that the contact area is not increased appreciably during the running-in period of the machine. When the full load and speed of the machine are required such surfaces may become overloaded locally and suffer severe damage. Yet, if the bearing surfaces are initially too rough, the initial wear will be high and correct operating clearances may be exceeded before the run-in is completed. When it is not practical to provide a definite running-in period on a machine, protective surface coatings, such as the phosphate type, are useful to allow rapid initial wear.

EDITOR'S NOTE: A companion article to this one, dealing with practical aspects of specifying surface finish, will appear in the September issue of PRODUCT ENGINEERING.

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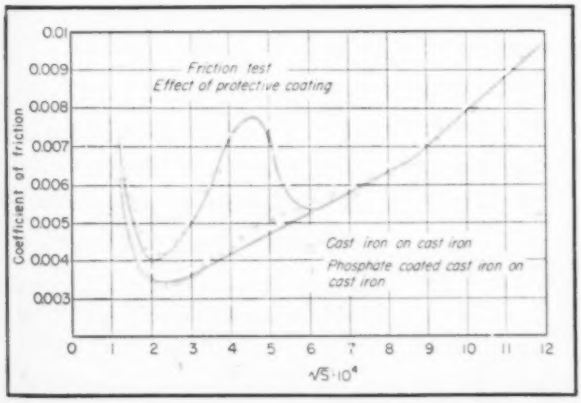


Fig. 8—Performance of uncoated cast iron surfaces, and of two surfaces, one of which is coated with a phosphate preparation.

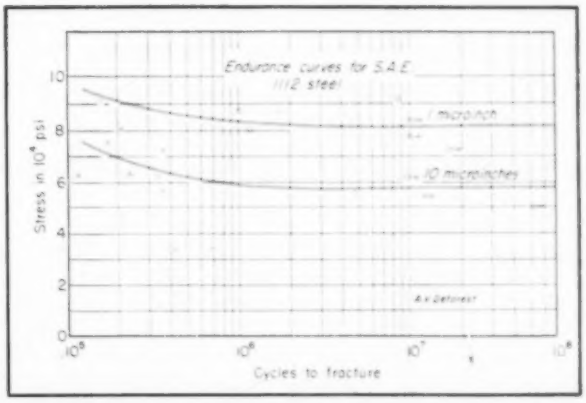


Fig. 9—Effect of surface finish on fatigue endurance limit for one sample of steel and two different surface conditions.

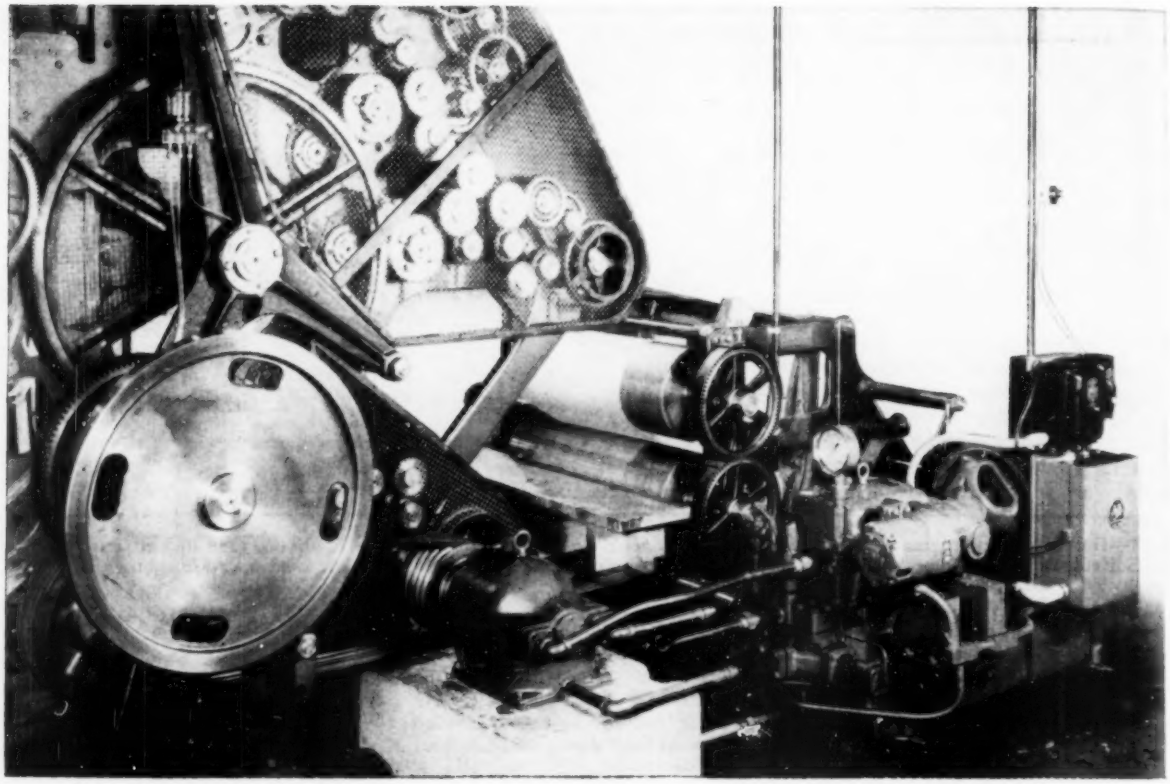


Fig. 1—One way variable delivery pump with electric pilot motor control and a constant displacement hydraulic motor with a V-belt drive applied to a roll bed printing press. Drive must be started, accelerated and stopped smoothly while under load.

Acceleration of Hydraulic

J. K. DOUGLAS, DEVELOPMENT ENGINEER, AND D. E. NILES, ENGINEER,
The Oilgear Company

Loading, friction, and motor characteristics that determine the time required by constant displacement hydraulic motors to produce a given change of speed in the driven machine. Method for calculating time required to accelerate and decelerate, and for computing distance travelled during speed change, can be applied to rotary and linear loads.

MACHINES THAT MUST BE STARTED or stopped under load, or those in which speed must be changed a definite amount from operation to operation in their work cycle, are frequently driven by constant displacement hydraulic motors. To select or design accurate adjustable pressure controls for the pumps that supply the motors in such applications, an analysis of pump pressure requirements at all portions of the work cycle is essential.

When the displacement of the motor per revolution and the pressure loss in the motor resulting from internal friction are known, it is not difficult to calculate the pressure that must be delivered to the motor for the different portions of the work cycle. When the time interval for making speed changes and the inertia of the masses in the driven system are also known, even the calculation of pressure required to accelerate and decelerate the drive and its driven system is readily accomplished by a simple logical procedure.

T_M = input torque to hydraulic motor, lb in.
T_L = torque required to balance friction losses in the motor, lb in.
T = torque delivered to motor shaft, lb in.

For equilibrium when the motor is driving the machine, Fig. 2(A), of course, the following relations exist.

$$T = T_M - T_L \quad (1)$$

Where

P = pressure differential across motor ports, psi

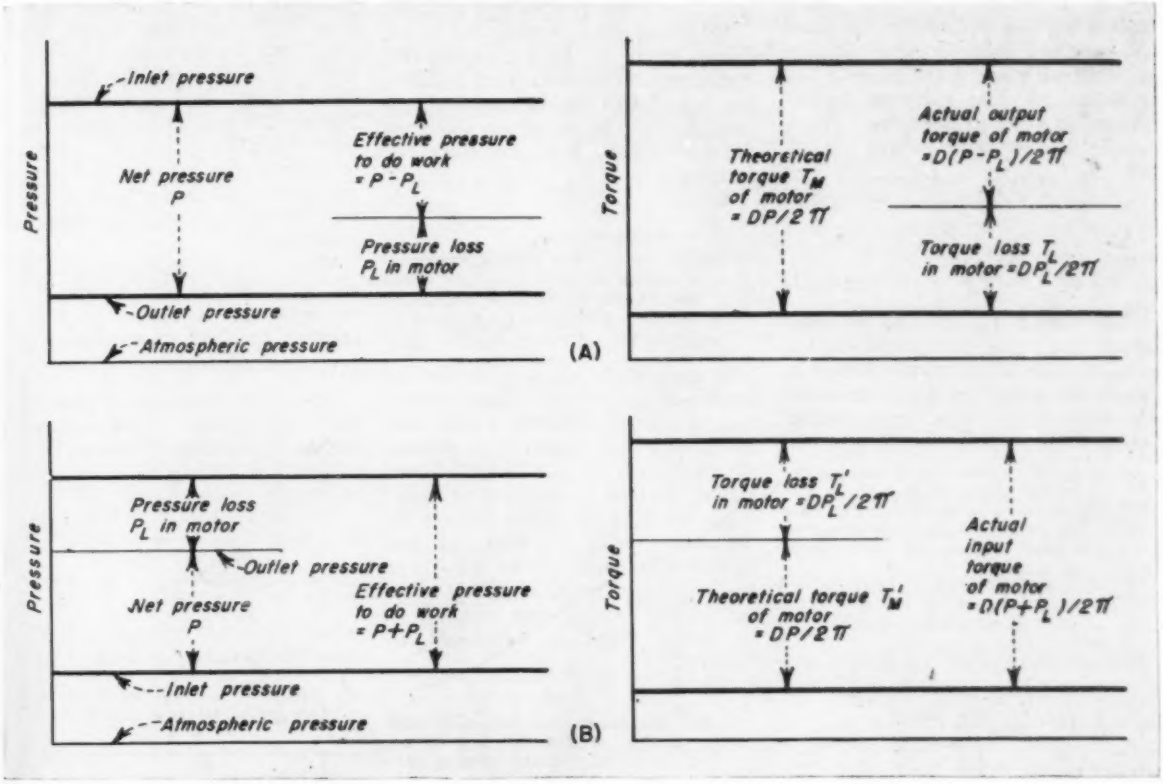

Pressure
Inlet pressure
Net pressure P
Effective pressure to do work = P - P_L
Pressure loss P_L in motor
Outlet pressure
Atmospheric pressure
(A)
Torque
Theoretical torque T_M of motor = DP/2π
Actual output torque of motor = D(P - P_L)/2π
Torque loss T_L in motor = DP_L/2π
Pressure
Pressure loss P_L in motor
Outlet pressure
Net pressure P
Effective pressure to do work = P + P_L
Inlet pressure
Atmospheric pressure
(B)
Torque
Torque loss T'_L in motor = DP_L/2π
Theoretical torque T'_M of motor = DP/2π
Actual input torque of motor = D(P + P_L)/2π

Fig. 2—Fundamental pressure and torque relations in the performance of a hydraulic motor. (A) Conditions while the hydraulic motor is accelerating a load or doing outside work. (B) Conditions while the hydraulic motor is decelerating a load.

Motors Driving Rotary and Linear Loads

P_L = pressure loss in motor resulting from internal friction, psi
D = displacement of motor per revolution, cu in.

the input torque T_M to the motor is

$$T_M = DP/2\pi \quad (2)$$

and the torque T_L required for motor friction losses is

$$T_L = DP_L/2\pi \quad (3)$$

Therefore from Eq (1) the output torque T at the motor shaft is

$$T = (DP/2\pi) - (DP_L/2\pi) = D(P - P_L)/2\pi \quad (4)$$

For conditions where the machine drives the motor, Fig. 2(B), so that the motor becomes a pump, the torque relations are changed.

Where

T' = torque delivered to motor by the machine, lb in.
T'_L = torque to balance friction losses in the motor, lb in.
T'_M = output torque of motor, lb in.

$$T' = T'_M + T'_L = D(P + P_L)/2\pi \quad (5)$$

The pressure loss P_L resulting from internal friction of the motor is a motor characteristic and must be determined experimentally. This pressure loss varies with speed and pressure but average values can be determined and used with reasonable accuracy.

To calculate the torque T on the driven shaft at the coupling to the motor, the external work done and the inertia effects resulting from several conditions of drive transmission and loading must be considered.

Torque T in Eq (1) is the sum of the torque required to do external work and, during speed changes, the torques required to accelerate or decelerate the masses that make up the driven system: where

T_D = torque at the motor shaft required to do external work, lb in.
T_A = torque available at the motor shaft to accelerate or decelerate masses in the driven system, lb in.

$$T = T_D + T_A \quad (6)$$

When the driven system is operating at constant speed the term T_A in Eq(6) is zero.

Where

P_d = increment of pressure required for external work load, psi

$$T_D = P_d D/2\pi \quad (7)$$

The torque T_D is that required to move the external or drag loads. These consist of friction loads and work output of the system. As far as drag loads are concerned, friction and work are

treated as similar quantities. Usually torque T_D can be readily calculated for most machines by a consideration of their work load and design.

When the drag loads can be expressed in terms of horsepower H_D, a simple relation exists; where

S = speed of motor, rpm
$H_D = P_d D\, S/12 \times 33{,}000$

or

$$P_d D = \frac{396{,}000\, H_D}{S} \tag{8}$$

The torque T_A required to make speed changes is determined from a consideration of the time interval in which the speed change is made and the inertia of the masses in the system, where

α = angular acceleration of motor shaft, radians per sec per sec
I = moment of inertia of the motor plus the effective moment of inertia of system referred to motor shaft speed, slug ft²
T_A = torque available for acceleration, lb in.
S_1 = initial motor speed, rpm
S_2 = final motor speed, rpm
t = time interval for speed change, sec

$$\alpha = \frac{T_A}{12I} = \frac{T - T_D}{12I} = \frac{2\pi}{60t}(S_2 - S_1)$$

or

$$t = \frac{24\,\pi\, I\,(S_2 - S_1)}{60\,(T - T_D)} = \frac{1.26\, I\,(S_2 - S_1)}{T - T_D} \tag{9}$$

or

$$T_A = T - T_D = 1.26\, I\,(S_2 - S_1)/t \tag{10}$$

by substituting Eqs (4) and (7) in Eq (9) there results

$$t = \frac{1.26\, I\,(S_2 - S_1)}{D\,(P - P_L - P_d)/2\pi} = \frac{7.9\, I\,(S_2 - S_1)}{D\,(P - P_L - P_d)} \tag{9A}$$

or

$$D\,(P - P_L - P_d) = 7.9\, I\,(S_2 - S_1)/t \tag{11}$$

The moment of inertia I in Eqs (9), (10), and (11), of course, is a composite term. It is the sum of the moments of inertia referred to motor shaft speed of the individual moving masses in the system and the moment of inertia of the motor.

When the motor is directly connected to the driven system without employing any speed changing elements, that is, when the motor and all parts of the driven system rotate at the same speed, I is determined as follows:

Where

I_m = moment of inertia of motor,
$I_1, I_2, I_3 \ldots I_n$ = moments of inertia of component masses 1, 2, 3, to n, respectively, in the driven system about their individual axes of rotation

$$I = I_m + I_1 + I_2 + I_3 \ldots + I_n \tag{12}$$

When the motor is connected to the driven system by speed changing elements, so that the motor speed is different than that of the driven system, or parts of the driven system rotate at different speeds, I is determined as follows:

Where

S = motor speed, rpm
$N_1, N_2, N_3 \ldots N_n$ = speed of revolution of component parts 1, 2, 3, to n, respectively in the driven system, rpm

$$I = I_m + I_1\,(N_1/S)^2 + I_2\,(N_2/S)^2 + I_3\,(N_3/S)^2 \ldots + I_n\,(N_n/S)^2 \tag{13}$$

In applications where an element in the driven system has a linear motion such as a weight lifted by a cable and drum, or a rack moved by a pinion, the inertia of the part having linear motion must be converted to an equivalent moment of inertia referred to the motor speed, and then included in the right hand side of Eq (13). The equivalent moment of inertia of the linearly moving part referred to the motor speed is determined as follows:

Where

R = radius of the pinion, gear or drum that is actuating the linearly moving part or load, ft
V = velocity of linearly moving part, ft per min
N_d = speed gear or drum, rpm
W = weight of linearly moving part or load, lb

$$\text{Equivalent } I = \frac{W R^2}{32.2}\left(\frac{N_d}{S}\right)^2 = \frac{W}{32.2}\left(\frac{V}{2\pi N_d}\right)\left(\frac{N_d}{S}\right) \tag{14}$$

Viscous Friction

The calculation of torque required to accelerate a load in which viscous friction must be overcome presents some difficulties. Viscous friction is a type of friction that varies directly with the relative speed between the elements and is independent of the forces involved. Actually, friction between all lubricated parts is of this type, but the effect of speed is generally neglected.

About the only instance in which viscous friction must be considered is when a viscous fluid is being "worked". An example is a wood pulp stirring machine in which a hydraulic motor drives a beater in a heavy wood pulp mixture. The beater must accelerate while in the mixture and the variation of viscous drag causes a variation of acceleration.

In problems where viscous friction occurs, the load torque characteristics take the form

$$T_v = KS \tag{15}$$

that is, the torque T_v necessary to overcome the viscous drag equals a constant times the motor speed S. For any particular application, the value of K can be determined experimentally by holding the motor speed constant and then calculating the torque T_v required to operate the viscous load at that speed. By substituting these values in Eq (15), the value of K can be readily computed.

When T is the total torque available, and T_v is the variable torque of the viscous drag, the torque T_x that accelerates the load at any instant is

$$T_x = T - T_v = T - KS \tag{16}$$

Since acceleration in any time interval is proportional to the accelerating torque during that interval.

$$\frac{2\pi}{60}\,\frac{dS}{dt} = \frac{T_x}{12\,I} = \frac{T - KS}{12\,I}$$

or

$$dt = \frac{I\,dS}{0.796\,(T - KS)} = \frac{I\,dS}{0.796\,[(T/K) - S]}$$

$$\int_0^t dt = \frac{I}{0.796\,K} \times \int_{S_1}^{S_2} -\frac{dS}{S - (T/K)}$$

$$t = \frac{I}{0.796\,K}\left\{\left[-\log_e (S_2 - T/K)\right] - \left[-\log_e (S_1 - T/K)\right]\right\} = \frac{2.89\, I}{K}\log_{10}\frac{T - KS_1}{T - KS_2} \tag{17}$$

The time required to accelerate from S_1 to S_2 can be determined from Eq (17). This equation does not apply to deceleration. Under decelerating conditions, Eq (16) becomes

$$T_x = T + T_v = T + KS$$

and the derived equation for time to decelerate is

$$t = \frac{2.89\, I}{K}\log_{10}\frac{T + KS_2}{T + KS_1} \tag{18}$$

This equation gives the time required to decelerate from S_1 to S_2. If the motor must reverse under viscous load, the solution procedure consists of solving two separate problems. First, calculate the time required to decelerate from S_1 to O rpm by Eq (18), and then add this time to the time required to accelerate from O to S_2 as determined by Eq (17).

If viscous friction is not present during a speed change, the average speed S_A is

$$S_A = (S_2 - S_1)/2, \text{ rpm} \tag{19}$$

and the revolutions made in t sec is

$$t\,S_A/60 = (S_2 - S_1)\,t/120 \tag{20}$$

If viscous friction is present, S_1 is difficult to calculate. By choosing the proper parameters, however, Eqs (17) and (18) can be plotted as shown in Fig. 3. These curves can be integrated graphically to give the curves shown in Fig. 4, which show revolutions of motor in elapsed time. If it is desired to know the revolutions made by the motor, determine t_1 and t_2 corresponding to S_1 and S_2 from Fig. 3, and then determine the revolutions made from t_1 to t_2 from Fig. 4. Incidentally, t_2

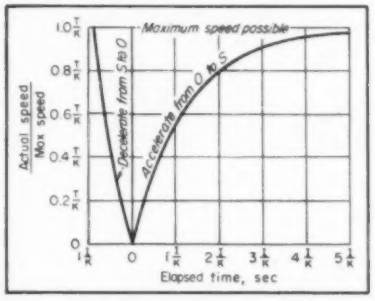


Fig. 3—Time to accelerate and decelerate a viscous friction load in terms of the ratio of actual speed to maximum possible speed and equation parameters.

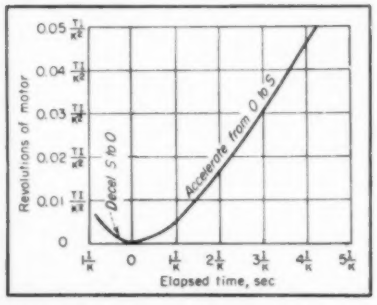


Fig. 4—Time to accelerate and decelerate a viscous friction load in terms of revolutions of motor and equation parameters.

minus t_1 is the time as calculated from Eqs (17) and (18) required for the speed change.

To illustrate the use of the equations presented in this discussion, three typical problems will be solved

PROBLEM 1. A hydraulic motor is directly connected to a machine having an inertia load referred to the motor shaft of 0.490 slug ft². A pump with a pressure actuated control supplies oil to the motor at a constant pressure of 1300 psi with a return pressure of 120 psi. The motor constants are

I_M = 0.043 slug ft² (moment of inertia of motor)
P_L = 60 psi (frictional loss in motor)
D = 4.83 cu in (displacement of motor per revolution)

If the machine does no external work and friction is negligible, how much time is required to accelerate to 1800 rpm from standstill?

SOLUTION 1. This problem deals with a simple inertia load and Eqs (9) and (9A) apply.

Since no drag load is present, P_d is zero.

$$P = 1300 - 120 = 1180 \text{ psi (pressure across motor ports)}$$
$$P - P_L = 1180 - 60 = 1120 \text{ (pressure available for acceleration)}$$
$$I = I_L + 0.490 = 0.533 \text{ slug ft}^2$$
$$S_2 - S_1 = 1800 - 0 = 1800$$
$$t = \frac{7.9 \times 0.533 \times 1800}{1120 \times 4.83} = 1.4 \text{ sec}$$

PROBLEM 2. A 3,000 lb reciprocating carriage is to be operated with a hydraulic motor by means of a chain and a 3 in. dia sprocket. A 4 to 1 reduction gear box, having a moment in inertia I_G of 0.542 slug ft² referred to the motor shaft, between the hydraulic motor and the sprocket shaft requires 50 lb in. of torque to overcome internal friction. The carriage has steel wheels running on steel tracks, the coefficient of friction is 0.007. Hydraulic motor constants are

I_M = 0.307 slug ft₂ (moment of inertia of motor)
P_L = 180 psi (frictional loss in motor)
D = 7.55 cu in. (displacement of motor per revolution)

If the line pressure is kept at 1600 psi and the return pressure at 35 psi, how much time is required for the carriage starting from standstill to reach a speed of 200 ft per min, and how far will the carriage travel before reaching that speed?

SOLUTION 2. This problem deals with linear motion, speed reduction and dry friction.

Using Eq (14), the ratio of speed of sprocket to speed of motor is

$$N_d/S = 1/4$$

and the equivalent moment of inertia I_L of the linear load is

$$I_L = \frac{3{,}000 \times (3/2 \times 12)^2}{32.2} \left(\frac{1}{4}\right)^2$$
$$= 0.091 \text{ slug ft}^2$$
$$I = I_M + I_G + I_L$$
$$= 0.307 + 0.542 + 0.091$$
$$= 0.940 \text{ slug ft}^2$$

Motor speed S_2 at a carriage speed of 200 ft per min is

$$S_2 = \frac{200 \times 4}{3\pi/12} = 1016 \text{ rpm}$$

The torque T_R at the motor required to overcome the friction of the carriage wheels on the rails is

$$T_R = \frac{3000 \times 0.007 \times 3}{4 \times 2} = 7.875 \text{ lb in.}$$

Total torque T_D required at the motor to do external work is

$$T_D = 50 + 7.875 = 57.875 \text{ lb in.}$$

from Eq (7)

$$P_d = 2\pi T_D/D = 2\pi \times 57.875/7.55 = 48.2$$

then from Eq (9A)

$$t = \frac{7.9 \times 0.940 \times (1016 - 0)}{7.55\,[(1600 - 35) - 180 - 48.2]} = 0.747 \text{ sec}$$

Average speed of carriage during acceleration is

$$200/60 \times 12 = 1.67 \text{ ft per sec}$$

and distance carriage travels during acceleration is

$$1.67 \times 0.747 = 1.25 \text{ ft}$$

PROBLEM 3. A pulpwood stirrer with a moment of inertia of 25 slug ft² requires 3600 lb in. of torque to operate at 900 rpm. A hydraulic motor connected directly to the stirrer has the following constants:

I_M = 2.74 slug ft² (moment of inertia of motor)
P_L = 200 psi (frictional loss in motor)
D = 20.05 cu in. (displacement of motor per revolution)

How much time is required to reach 900 rpm with a line pressure of 1700 psi and a 35 psi return pressure?

SOLUTION 3. This problem deals with viscous friction and Eq (15) applies.

$$K = 3{,}600/900 = 4 \text{ lb in. per rpm}$$
$$I = 25 + 2.74 = 27.74 \text{ slug ft}^2$$
$$P = 1700 - 35 = 1665 \text{ psi}$$

Motor output torque from Eq (4) is

$$T = 20.05\,(1665 - 200)/2\pi = 4{,}670 \text{ lb in.}$$

Substituting in Eq (17)

$$t = \frac{2.89 \times 25}{4} \log_{10} \frac{4{,}670 - 0}{4{,}670 - 3{,}600}$$
$$= 18.06 \log_{10} 4.36 = 18.06 \times 0.6395$$
$$= 11.55 \text{ sec}$$

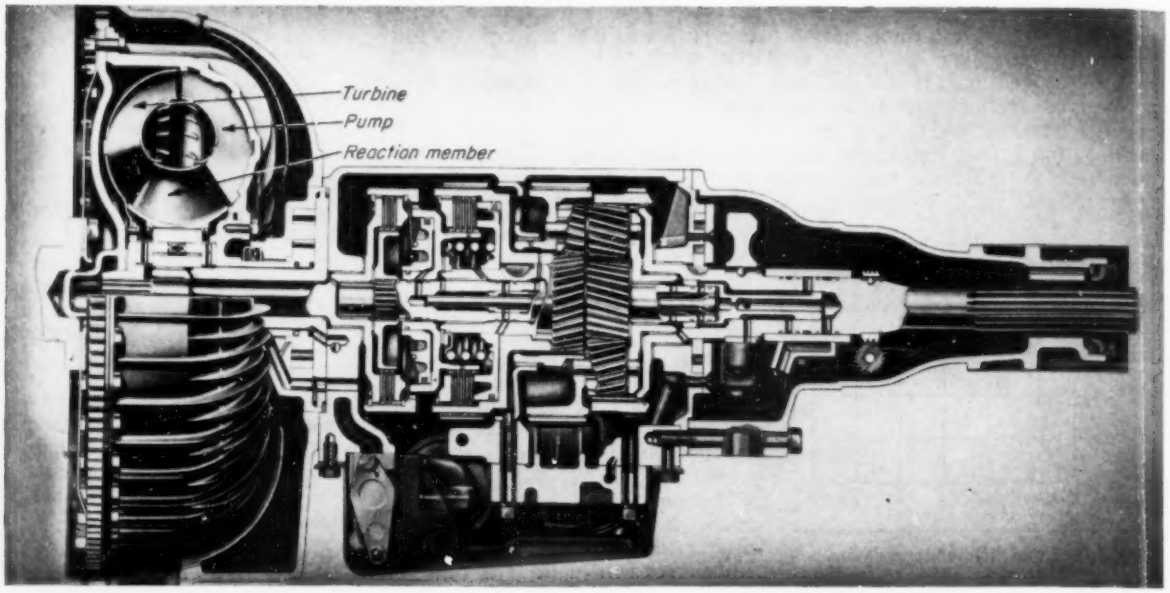
Time t can also be determined with the aid of Fig. 3. For

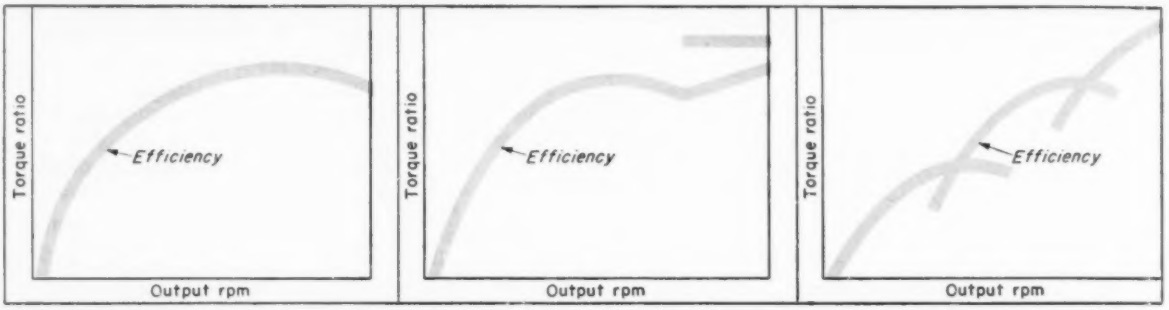
$1.0\ T/K = 4{,}670/4 = 1{,}168$ rpm (max speed possible)
$S/S_{max} = 900/1{,}168 = 0.771$

The elapsed time corresponding to 0.771 T/K when accelerating is 1.85 I/K or

$$t = 1.85 \times 25/4 = 11.56 \text{ sec}$$

CONCLUSION. With the help of the equations derived in this article the designer, of course, can with little difficulty determine the net pressure needed across the motor ports to accelerate a machine in a desired time, when drag loads, inertia loads, and viscous loads are known.



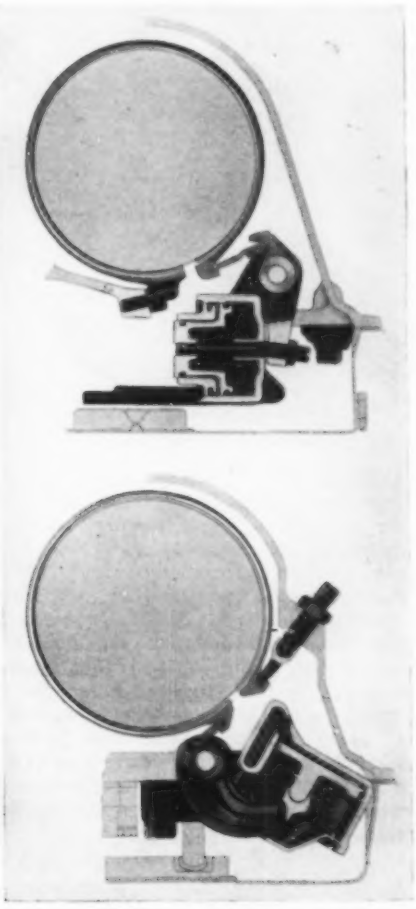


EXAGGERATED EFFICIENCY CURVES illustrate three basic ways of using a single stage torque converter. In curve (1) the converter is used alone. A free wheeling reaction member and, or a secondary pump may be used to raise efficiency. In curve (2) a lock-up clutch is engaged to give full mechanical transmission at high speeds. The Ford-Mercury choice (3), uses an automatic three-speed mechanical transmission behind the torque converter.

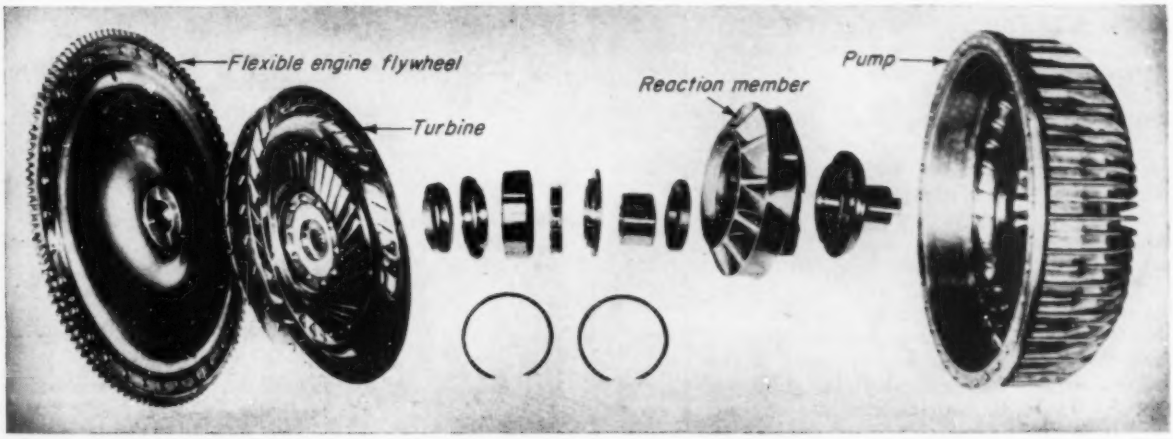
New Ford Automatic Transmission

Designed for use with both Ford and Mercury automobiles, this latest development in automatic transmissions is a joint product of the Borg-Warner Co., and the Ford Motor Co. The new system uses a single stage fluid torque converter of the simplest type, having only the three basic elements needed for torque multiplication. Connected to the torque converter is a three-speed mechanical transmission which changes gear ratios automatically to take advantage of top converter efficiency. The gear box is of the planetary type to permit shifting under torque. It has a gear set of the double pinion type, with three sets of long and short pinions and a ring gear. This total of nine gears provides three forward ratios, two geared and one direct, and one reverse ratio. Two oil pumps, one driven at all times by the engine and one by the output shaft, deliver oil under pressure to charge the converter, actuate and control bands and clutches, and pressure lubricate the entire mechanism. The transmission control system consists of a pressure regulator assembly, a hydraulic governor mounted on the output shaft, and the main hydraulic control unit or valve body assembly where all of the manual and automatic control functions are centered. Unlike several other manufacturers, Ford has not found it necessary to increase the power of its standard engine because of the automatic transmission. This will answer one of the main arguments against the use of automatic transmissions in automobiles.

CONVERTER AND TRANSMISSION OIL are cooled by air circulation through the converter housing. Fins cast on the pump cover serve the dual purpose of providing cooling surfaces and vanes for pumping cooling air. Air is drawn in at bottom left side, passes through and is exhausted.



TWO MULTIPLE DISK CLUTCHES and two brake bands (above) determine power flow. Front brake band and servo mechanism; top, rear brake, bottom.



TORQUE CONVERTER ASSEMBLY contains only the three basic elements needed. Pump has a die-cast aluminum cover and 31 stamped steel blades. The turbine assembly is made of steel stampings, except for a forged hub. The reaction member, or stator, is a die-casting.

(*continued on next page*)

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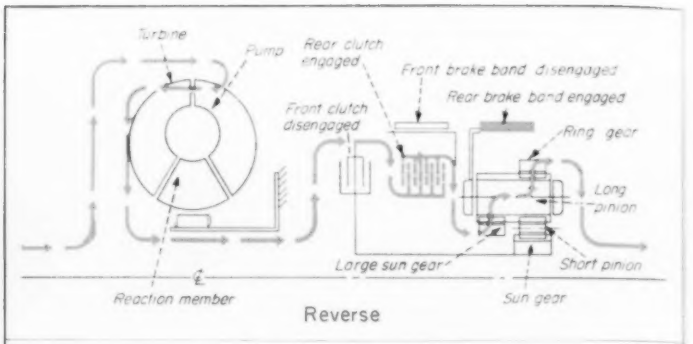
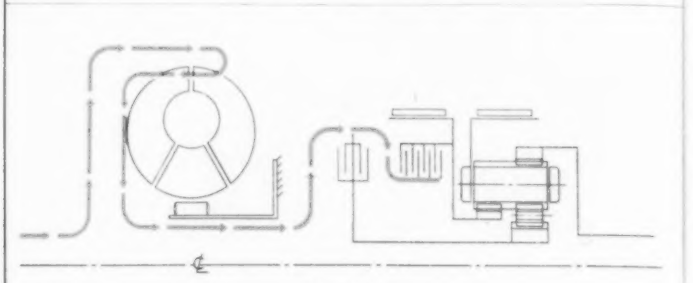
Ford (continued)

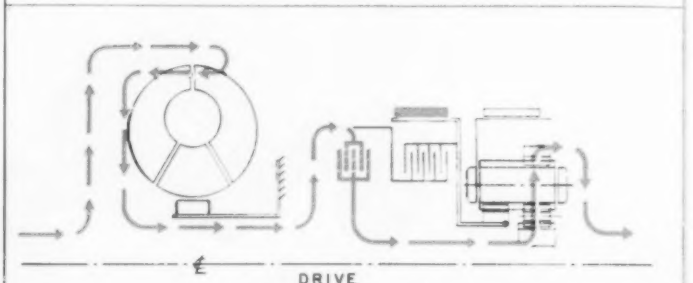
DRIVER CONTROLS are located on the steering column. Positions on the selector quadrant are arranged slightly differently than on other automatic transmissions. From left to right (see illustration above) they are the Park position, Reverse, Neutral, Drive, and Low. In the Park position a pawl engages teeth on the periphery of the internal gear and locks the driven shaft against rotation. The Drive position includes both High and Intermediate gear. While in the Drive position the High ratio (1 to 1) and Intermediate ratio (1.48 to 1) function automatically. Starting is ordinarily done automatically in the Intermediate range, which has a 1.48 to 1 ratio, assisted by the torque converter which has a ratio of 2.1 to 1, or a total equivalent of a 3.11 to 1 ratio. This range is comparable to low gear in a standard transmission. The shift to High is automatically made at speeds between 17 and 63 mph depending on the degree of throttle opening. Below 40 mph a manual shift to the Low range places the transmission in a 2.44 to 1 ratio. Above 40 mph a manual shift to Low down-shifts the transmission to Intermediate range, where it remains regardless of throttle opening. Shifting between Low and Drive is possible at any throttle opening.

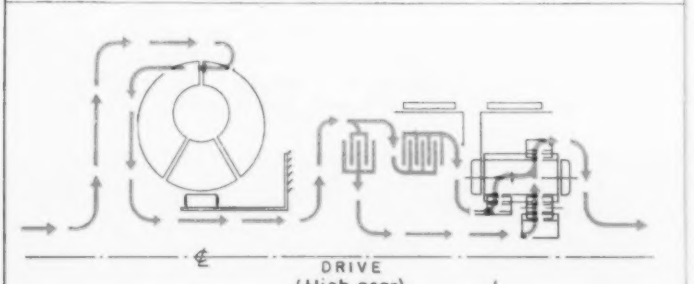
Power flow is explained (right) by schematic diagrams. In Reverse, the pinion cage is held against rotation by the rear brake band and drive is from the large sun gear across the long pinion, to the ring gear. This causes a reversal in direction between input and output shafts. In Neutral, both clutches and bands are disengaged. In Intermediate, the front clutch is engaged and the forward sun gear is coupled to the input shaft of the transmission, the front band holds the large sun gear against rotation and the pinion cage is free to rotate in the same direction as the input shaft, but at a lower speed. In High, both clutches are engaged and both bands released. All parts rotate together to give a direct drive. Low gear is much the same as Intermediate, except that the rear band holds the pinion cage.

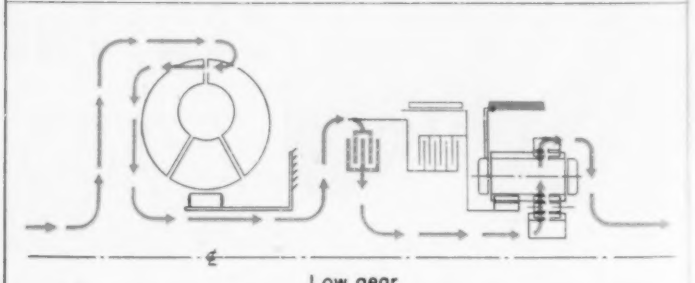


Reverse


Neutral


DRIVE
(Intermediate gear)

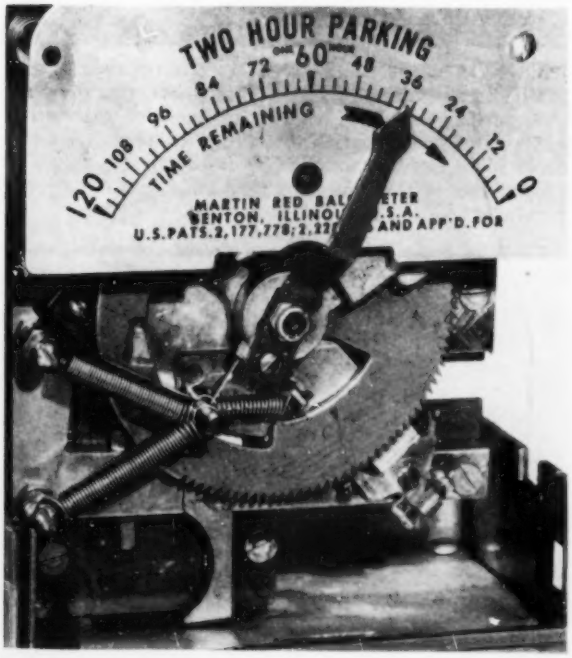

DRIVE
(High gear)


Low gear

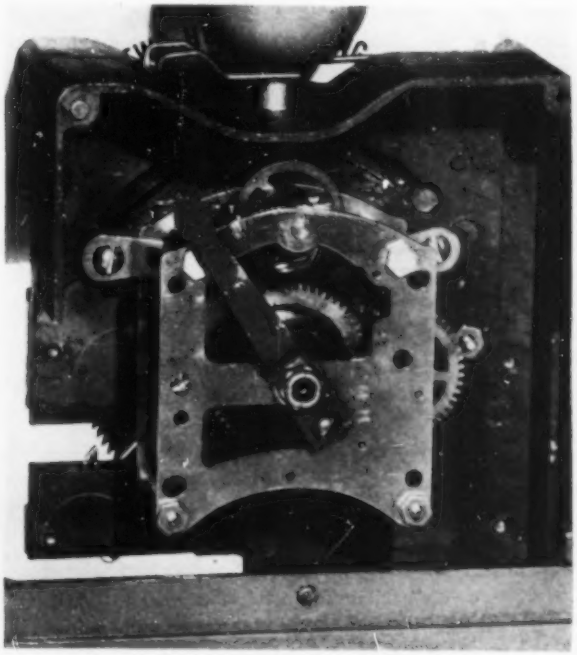
Die Castings Simplify Meter Design

The Martin Red Ball parking meter uses nine zinc die castings in the body design. The company's chief engineer gives these reasons for this extensive use: (1) dimensions are held within very close limits, (2) practically all machining is eliminated except drilling and tapping, (3) the process permits rapid production, and (4) gives a clean as-cast finish. The product's most important feature is the red ball which pops up to show a violation. The ball is easily visible at a distance. The new design is claimed to be especially tamper-proof and impervious to weather. The Red Ball Parking Meter Co., of Benton, Ill., is a division of the J. W. MenHall Drilling Company.

LEVERING COIN into the machine winds the clock mechanism and sets the correct amount of time on the dial. The meter needs no periodic winding as is common with many meters. Notice how few zinc die castings are needed to fabricate the entire parking meter shell.



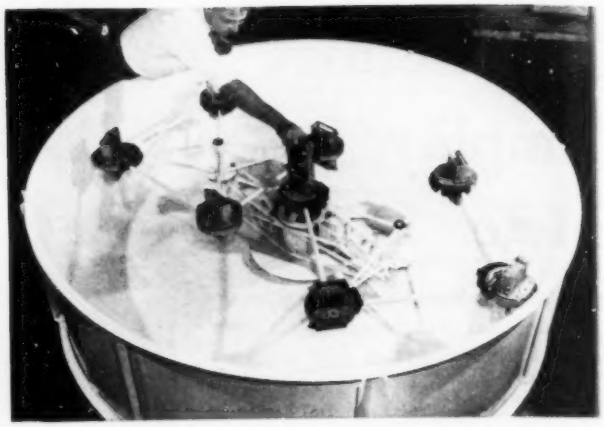
WORKING PARTS are of stainless steel or cadmium and chrome plated cold rolled steel. These materials play an important part in all-weather operation. Under conditions of high humidity followed by extreme cold, operation of the coin lever will clear the ice and allow normal operation.

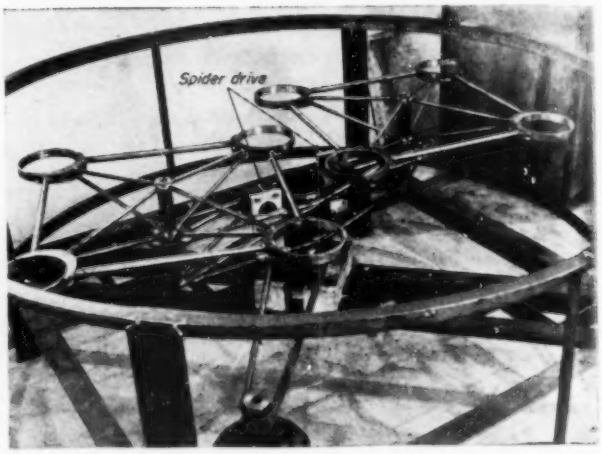


CLOCK MECHANISM operates at temperatures between −25 F and +120 F with an error not exceeding two minutes an hour. Only a fraction of the positive spring wind is used in operation of the meter, thus assuring uniform tension. Clock operates only when meter is in use.

Centrifuge Loads and Unloads At Full Speed

An experimental machine made especially for the food processing field, the Georgia Tech continuous centrifuge can be loaded and unloaded while the machine travels at top speed. The radius of gyration of each individual food container moves from zero up to maximum and then back to zero for unloading. Thus each container travels outward in a spiral and returns, with the loading and unloading taking place when the container holder axis is on dead center, exactly above the central axis. In the experimental machine the rate of discharge can be varied up to a maximum of three one-lb containers a minute. Each container is whirled for 3 minutes while being subjected to an increasing centrifugal force that reaches 334 g, maximum. During the stationary period, when a container is at dead center, the wet material, which has been previously packaged in a screened container, is inserted in the holder. The dry container is released at the same time and is pushed out through a discharge chute located in the hollow central axis. A production model of this experimental machine could have countless innovations. The machine was made in the Georgia Institute of Technology's Engineering Experiment Station, which sponsored the project in conjunction with the Tennessee Valley Authority.



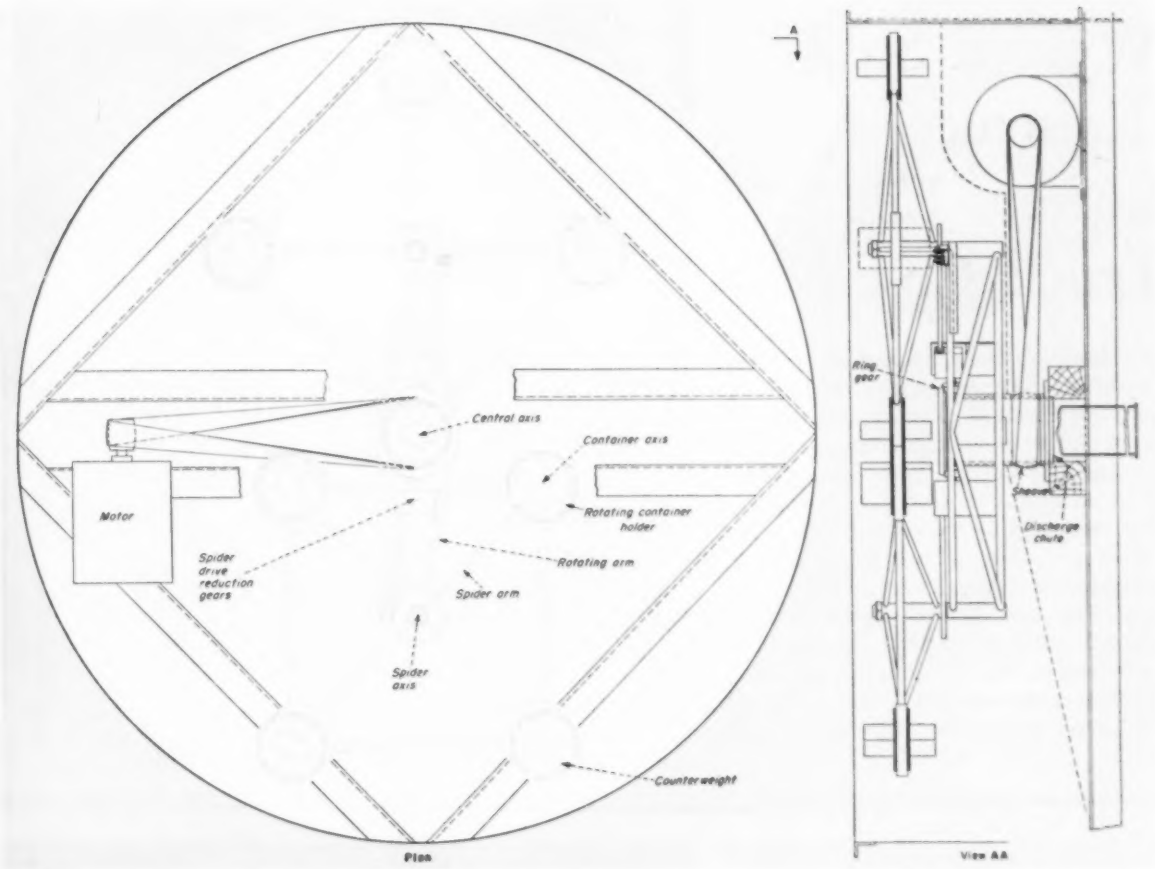


LOADING AND UNLOADING are accomplished simultaneously as each container comes into dead center position. Spider arms are rotated by a system of reduction gears driven from the central ring gear. The entire machine is driven by a variable speed electric motor which is belted to a sheave on the lower end of the central rotating arm.

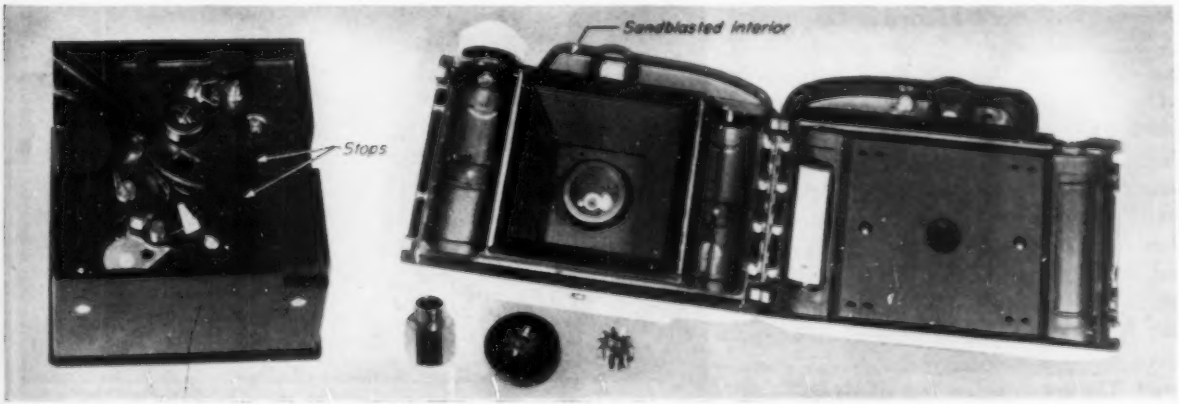
Redesign Cuts Assembly Time

ENGINEERS of the Whitehouse Products Co., in Brooklyn, N. Y. redesigned the Beacon Camera first from the viewpoint of appearance, and second with an eye to manufacturing economies. Expensive screw machined parts were replaced with stampings. Stops and fittings were molded in the plastic case wherever possible. Eyelets were extensively used and all unnecessary parts left out. The new design includes a coated double lens system, and a shutter designed for flash synchronization.

REDESIGNED CAMERA and accessories have well styled lines, yet are low-cost products. Adjustments have box-camera simplicity to appeal to the novice photographer.



FOUR CONTAINER HOLDERS are free to rotate about their own axis on each of two spiders. These two spiders are rotated slowly about the spider axis, which in turn revolves rapidly about the central axis. Since the length of each spider arm is exactly equal to the length of the radius arm connecting the spider to the central axis, the radius of revolution of any container holder about the central axis will vary continuously between a minimum of zero and a maximum of twice the length of the spider arm when the spider has rotated 180 deg from the zero position. Each container holder thus travels out from dead center and returns on a spiral path.



SHUTTER SYSTEM must conduct small electric current to facilitate flash synchronization. Parts are largely stamping of brass, nickel-silver, or phosphorous-bronze. Eyelets fasten all parts. To reduce glare without a painting operation, inside of camera is sandblasted while still in the mold. Shutter pinion is cut from a cold-drawn section.

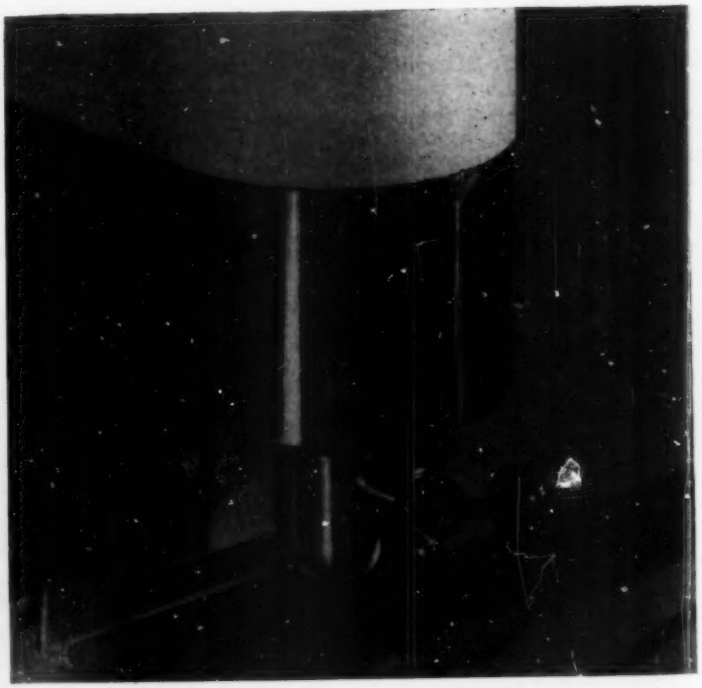
Welded Steel Construction Lowers Lift Truck Cost

By redesigning two parts of their lift truck for welded steel construction, the Mobilift Corp., of Portland, Oregon, reduced costs by 20 and 30 percent, respectively, and at the same time reduced weight without sacrificing strength. In both cases durability tests conducted by the company show the welded structures to be stronger and less susceptible to damage under adverse conditions. The redesigned parts were a combination flywheel and fan, and a fender. Both parts were previously made of cast iron. Welding equipment used is the product of the Lincoln Electric Co., Cleveland, Ohio.

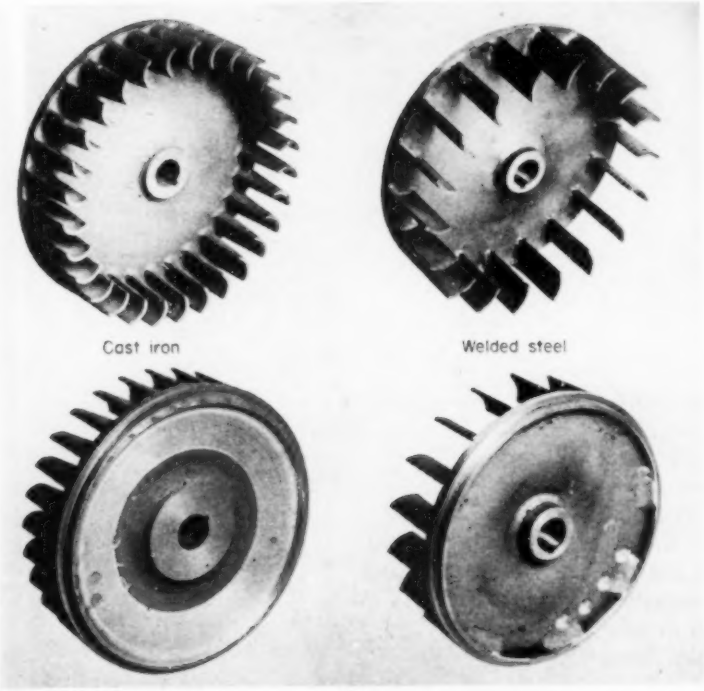
STEEL SIDE FRAME SECTION of truck was previously assembled with a cast iron fender. Redesigned welded steel fender saved 30 percent in cost, and reduced weight without loss of strength.

Stainless Steel Valve Meters Mixed Flow

The Canada Dry "In-Fountain Dispenser" is an especially designed beverage dispenser for use within the conventional syrup well of soda fountains. The most important feature, from a design standpoint, is the all stainless steel mixing valve and dispenser assembly made by the Hartford Machine Screw Co., Hartford, Conn. This screw machine-made valve automatically mixes soft drinks of any size between 5 to 12 oz, with guaranteed uniformity of product. No icing is required, since the dispenser derives its cold from the fountain syrup well. The construction complies fully with the most recent sanitary code rulings. The plastic syrup tank and cover are molded by the General Electric Company.

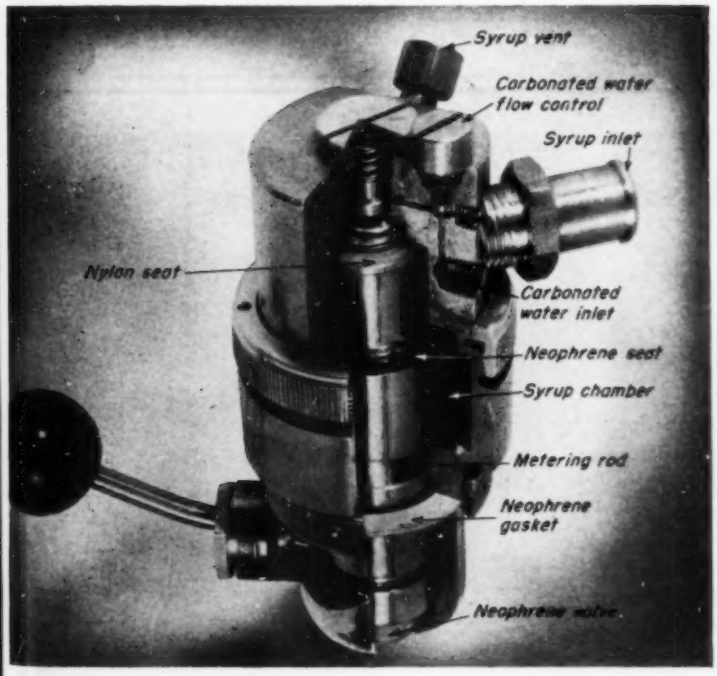


MIXING VALVE is mounted in counter-type syrup tank. Mixed drinks of consistently correct proportions is obtained by merely pressing the lever. Mixing valve can be adjusted for different mixtures and syrup viscosities as well as for different serving amounts.

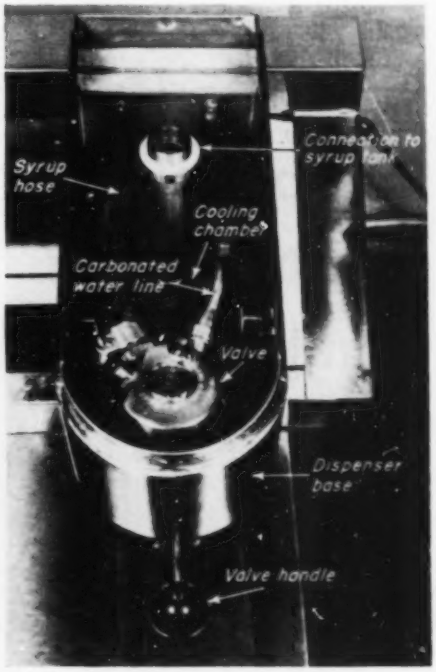


CONTRARY TO NORMAL PRACTICE, the objective was to create a lighter weight flywheel. Welded design reduced weight from 40 lb to 24, and also developed about 10 percent greater cooling capacity.

REDESIGNED welded steel fender, above, has good appearance and light weight with no apparent sacrifice in strength.



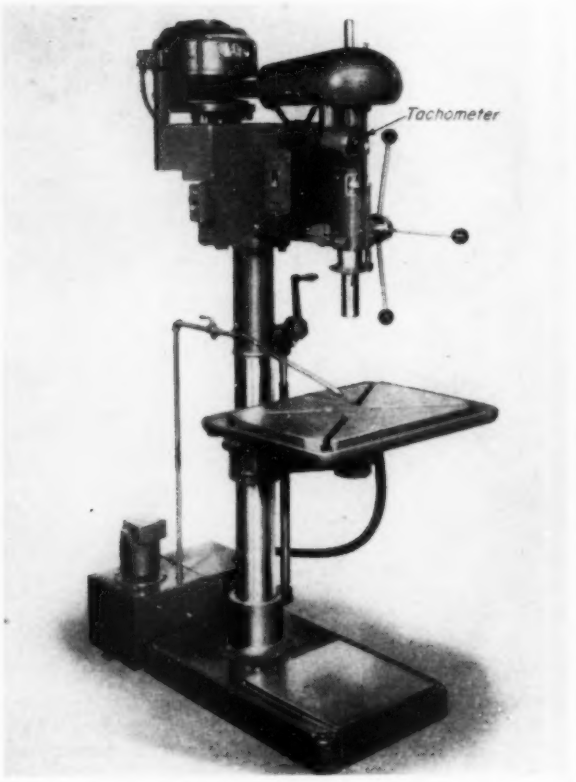
FOR OPERATION, lever raises interior assembly. This opens Neophrene valve plug at bottom and allows measured amount of syrup to flow. At same time upper Neophrene seat prevents the flow of more syrup into the syrup chamber. Carbonated water flows until serving glass is full.



SYRUP FLOWS from main tank into 25 oz stainless steel cooling chamber. No further icing is required. Entire mixing valve is screw machine made from stainless steel.

Drilling Machine Has Variable Speed Feature

The Model E-25 drilling machine, made by the Sibley Machine and Foundry Corp., South Bend, Ind., has several design features unique in its size and price range. Weighing only 765 lb, the machine has a 25 in. swing and a table working surface of 15 by 23 inches. The most unusual feature in a machine of this size is the infinitely variable spindle speed control. In the Model E-25 a standard Reeves Vari-Speed pulley is incorporated with a special Sibley adaption to give stepless speeds between motor and pulley limits. Two speed options are offered for any motor by the use of different sized spindle pulleys. The machine can be powered by 1,800, 1,200 or 900 rpm motors with spindle speed options ranging from 2,160 to 206 rpm. The variable speed drive permits speed selection within each option range. A Stewart-Warner tachometer, mounted at the front of the machine and geared to the spindle, visibly indicates the exact spindle speed.

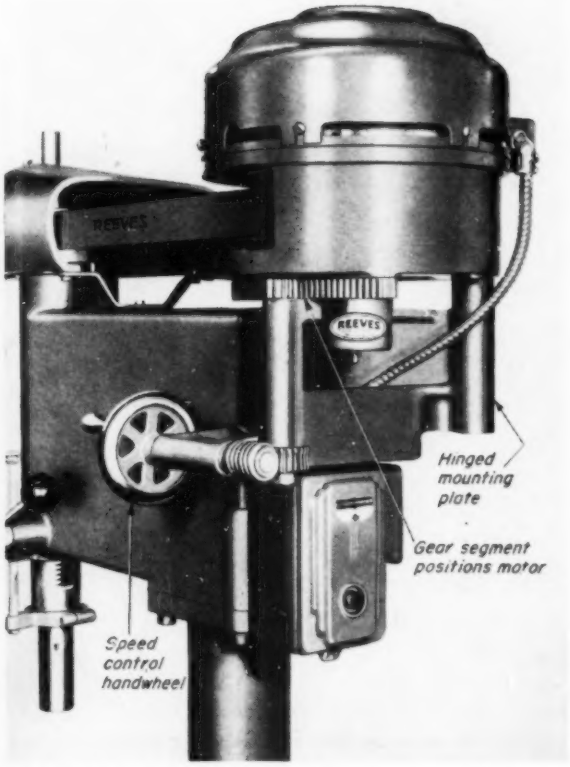


TACHOMETER keeps a constant check on drill speeds. Operator can quickly set correct spindle speed for each drill size. The large swing of 25 in. is unusual for a drilling machine of this size and price range.

Combination Lock Features Silent Tumblers

Ordinarily, the surest way of picking a combination lock is to hear or feel the tumblers drop into place. Older safe crackers sand-papered their fingertips to make them sensitive to feel. More scientific burglars have used everything from the stethoscope to supersonic hearing devices. Now Sargent & Greenleaf, Inc., of Rochester, N. Y., has developed a "Manipulation Proof" combination lock that makes absolutely no telltale click when tumblers fall. This feature is obtained by having a manganese bronze lever ride noiselessly on the circumference of the drive wheel, thus keeping the lever fence at least 0.030 in. away from the notched wheel tumblers.

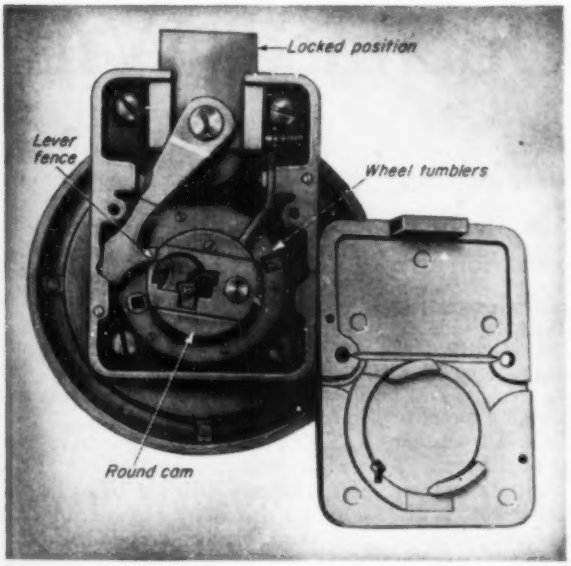
THE "Manipulation Proof" combination lock is either knob operated (above) or key operated. With key operation one user may possess key and another the combination.



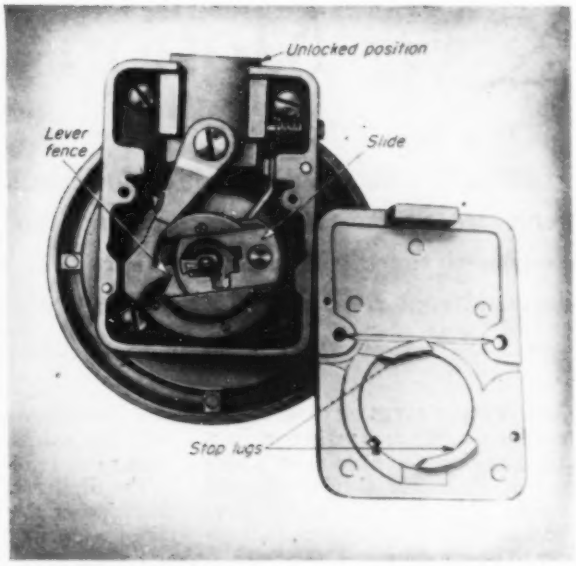
CONTROL HANDWHEEL and worm turn the gear segment, which moves motor about hinged section. A Reeves Vari-Speed pulley mounts directly on a 1½ hp Fairbanks-Morse Axial air gap motor for variable speed control.

Nylon Movement for Pressure Gage

THE NYLON-ALUMINUM PRESSURE GAGE MOVEMENT developed by Manning, Maxwell & Moore, Inc., of Bridgeport, Conn., was designed to reduce mass, and thereby lessen impact effects and increase sensitivity. The nylon absorbs impact and is particularly corrosion resistant. Aluminum provides the necessary lightness in mating parts to reduce impact and increase sensitivity.



LOCK WITH INNER SPINDLE is departure from usual practice. The lever fence rides on round cam at rim of the wheel tumbler. There is 0.030 in. clearance between them.



WHEN TUMBLERS have been soundlessly aligned, slide can be moved with key or knob. Lever fence then drops into the engaged position and retracts the bolt.

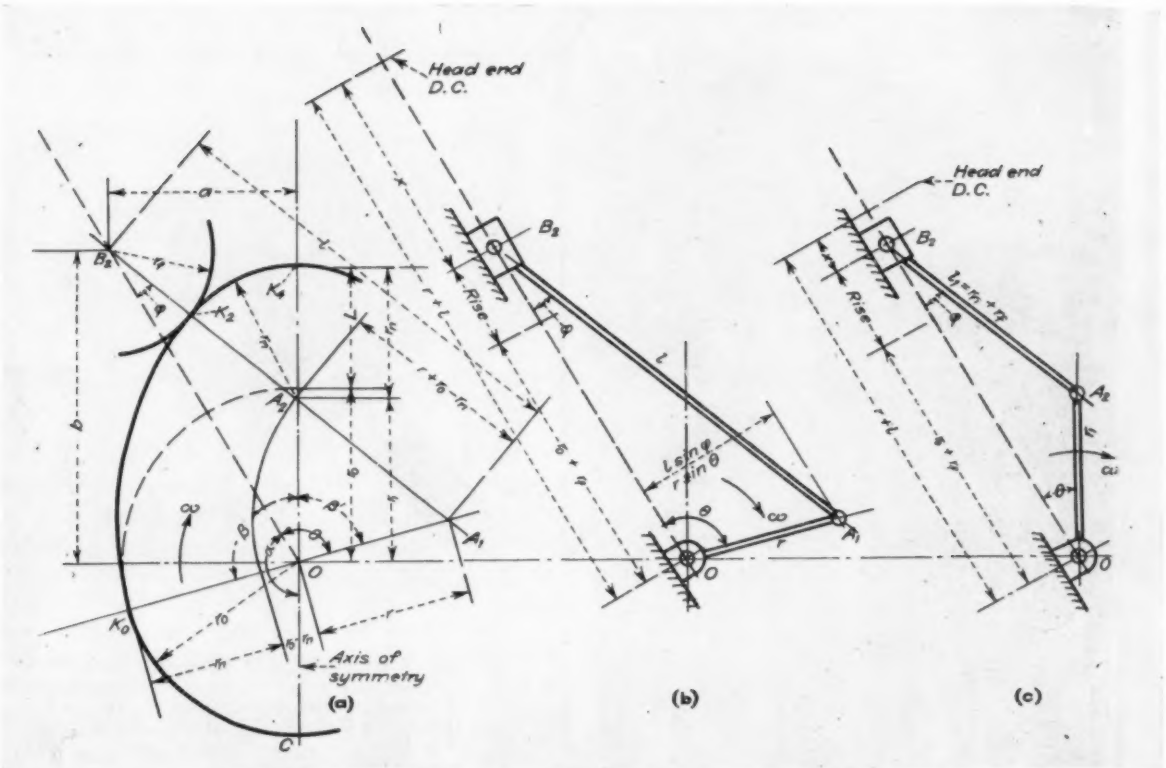


Fig. 1—(A) Cam composed of circular arcs with roller follower. Cam profile is symmetrical about the vertical centerline. (B) Equivalent slider-crank linkage when follower is on arc K_1K_2. (C) Equivalent slider-crank linkage when follower is on nose of cam.

Straight Line Follower Motions Obtained With

Plate cams with profiles made up of circular arcs and tangents are easiest to manufacture and can often be used. Equations for displacement, velocity and acceleration are derived from the kinematic similarity of the slider-crank and Scotch yoke with these types of cams and followers.

OF THE MANY TYPES OF CAMS used to transform rotary motion into reciprocating motion, the plate cams having arcs of circles, or circular arcs and straight lines, for their profiles have the advantage of being comparatively easy to manufacture.

The axis of rotation of plate cams is normal to the plate. The motion of the follower is in the plane of the cam. The follower goes through a complete cycle and returns to its initial position during one revolution of the cam.

The displacement, velocity and acceleration of the follower that is obtained with any type of cam, depends on the shape, size and angular velocity of the cam and the type of follower.

Since cams can be made in various shapes, they can be designed to impart many different types of motion to the follower. Complex motions can be obtained that might not be possible by other means. In general, designers are interested, not only in displacement of the follower, but also in its velocity and acceleration. In this article, only cams composed of arcs of circles and straight lines and the motions they impart to roller and to mushroom followers, will be considered.

Since the motion of followers actuated by cams composed of arcs of circles is kinematically similar to that of a piston driven by a connecting rod and crank mechanism, design equations for displacement, velocity and acceleration can be readily determined. These equations are also an aid in establishing the dimensions of such cams

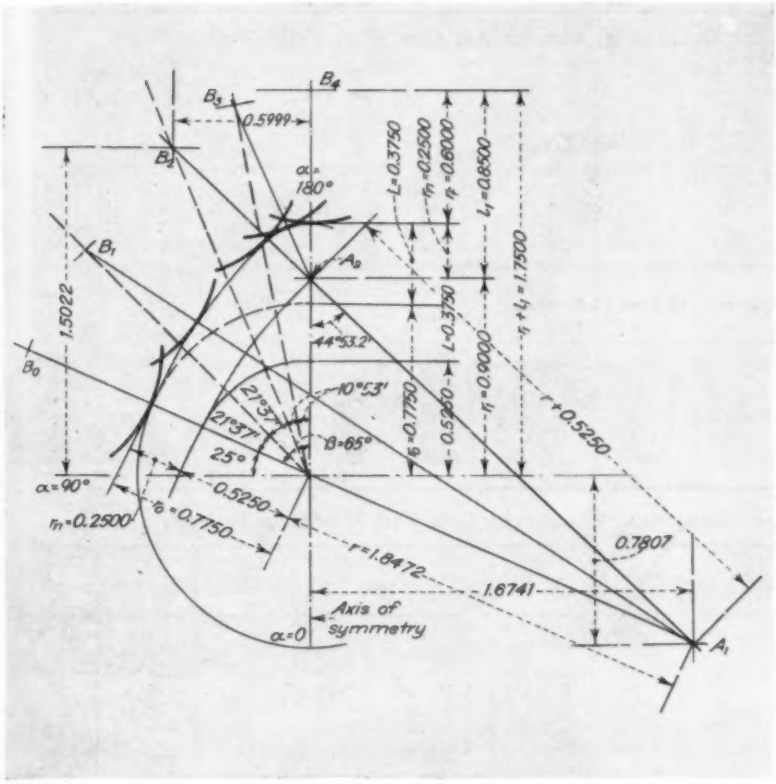


Fig. 2—Layout for cam of Example 1 with roller follower. Cam has a lift of 0.375 in. for a shaft rotation of 65 degrees. The radius of the follower roller is 0.600 inch.

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Mechanical Engineering Department,
Northwestern University

Circular Arc Cams

to obtain a necessary desired motion.

In Fig. 1(A) is shown the left-hand side of a cam profile that is symmetrical about the vertical centerline. The side shown is made up of three circular arcs with centers at points O, A_1 and A_2. The total rise L of the follower takes place during the rotation of the cam through the angle β. The roller follower is not offset.

To analyze the motion of the follower, assume that the cam is stationary and that the center line of the follower is rotating at the given angular velocity in the direction opposite to that of the actual rotation of the cam.

Over arc CK_o, of radius r_o with center at the axis of rotaton O, the follower receives no displacement, therefore, the values for velocity and acceleration of the follower are zero.

For all points of contact of the follower on arc K_oK_2 with the center at A_1, the distance from A_1 to the center of the roller is constant. Distance OA_1 is, of course, constant also. The kinematic relationship between the cam and follower is the same as that for a crank and piston in a slidercrank mechanism. This similarity is apparent from Fig. 1(B). When the roller makes contact at any point on arc K_oK_2, the length of the crank r is OA_1, and the length l of the connecting rod is radius A_1K_o plus radius r_r of the roller. Mathematical expressions for the displacement, velocity, and acceleration of the piston of a slider crank mechanism thus apply to those for a follower, without offset, that receives its motion from a cam composed of circular arcs.

Similar reasoning applies to arc K_2K_4. The equivalent slider-crank linkage is shown in Fig. 1(C). Here the crank r_1 is OA_2, and the connecting rod length l_1 is equal to radius A_1K_2, or r_n, plus the radius r_r of the roller.

Expressions for displacement, velocity, and acceleration are readily derived. Take the origin of coordinates at the head end dead center, that is, r plus l from the crank center.

Where

$\omega = d\theta/dt$ = constant angular velocity of crank, deg per sec
r = crank radius, in.
l = length of connecting rod, in.
$m = r/l$ = ratio of crank radius to length of connecting rod
θ = angle between the crank and the base line OB, deg
ϕ = angle between the connecting rod and the base line OB, deg
x = displacement of piston from head end dead center, in.

$$x = l + r - r\cos\theta - l\cos\phi \qquad (1)$$

Differentiation of Eq (1) with respect to time t gives the velocity $\dot{x}$ of the piston,

$$\dot{x} = r\,\omega\sin\theta + l\sin\phi\,\frac{d\phi}{dt} \qquad (2)$$

$$= r\left[\omega\sin\theta + (1/m)\sin\phi\,\frac{d\phi}{dt}\right] \qquad (2A)$$

Inspection of Fig. 1(B) shows that

$$l\sin\phi = r\sin\theta$$

or $$\sin\phi = m\sin\theta$$

or $$\phi = \arcsin(m\sin\theta) \qquad (3)$$

Differentiating Eq (3) with respect to θ

$$\frac{d\phi}{d\theta} = \frac{m\cos\theta}{\sqrt{1 - m^2\sin^2\theta}}$$

$$= \frac{m\cos\theta}{\sqrt{1 - \sin^2\phi}} = \frac{m\cos\theta}{\cos\phi} \qquad (4)$$

Hence

$$\frac{d\phi}{dt} = \frac{d\theta}{dt}\cdot\frac{d\phi}{d\theta} = \frac{\omega\, m\cos\theta}{\cos\phi} \qquad (5)$$

Substituting Eq (5) in Eq (2A)

$$\dot{x} = r\,\omega\,[\sin\theta + \cos\theta\tan\phi] \qquad (6)$$

In like manner, the acceleration $\ddot{x}$ of the piston is found by differentiating Eq (6) with respect to t,

$$\ddot{x} = r\,\omega^2\left[\cos\theta + \frac{m\cos^2\theta}{\cos^3\phi} - \sin\theta\tan\phi\right] \qquad (7)$$

Substitution of numerical values can be readily made in Eqs (1), (6) and (7) and the resulting values plotted.

When the piston is at the head end dead center, as for contact at K_4 in Fig. 1, both θ and ϕ have the value zero. Eqs (6) and (7) then become

$$\dot{x} = 0; \quad \ddot{x} = r\,\omega^2\,[1 + (r/l)] \qquad (8)$$

When the piston is at the crank end

Table I—Computations for Example 1. Circular Arc Cam With Roller Follower

Point	α	θ	r	l	r/l	$r\omega$	$r\omega^2$	$\sin\theta$	$\cos\theta$	$r\cos\theta$	$\frac{r}{l}\sin\theta = \sin\phi$
B_0	115°0′	180°0′	1.8472	3.2222	0.57327	174.10	16,408	0	−1.00000	−1.8472	0
B_1	136°37′	158°23′	1.8472	3.2222	0.57327	174.10	16,408	0.36839	−0.92967	−1.7173	0.21119
B_2	158°14′	136°46′	1.8472	3.2222	0.57327	174.10	16,408	0.68497	−0.72857	−1.3458	0.39267
B_3	158°14′	21°46′	0.9000	0.8500	1.05882	84.82	7,994	0.37083	0.92870	0.8358	0.39264
B_4	169°7′	10°53′	0.9000	0.8500	1.05882	84.82	7,994	0.18881	0.98201	0.8838	0.19992
B_5	180°0′	0	0.9000	0.8500	1.05882	84.82	7,994	0	1.00000	0.9000	0

Point	ϕ	$\cos\phi$	$l\cos\phi$	$\tan\phi$	$\sin\theta\tan\phi$	$\cos\theta\tan\phi$	$\frac{m\cos^2\theta}{\cos^3\phi}$	x	Rise	$\dot{x}$	$\ddot{x}$
B_0	0	1.00000	3.2222	0	0	0	0.5733	3.6944	0	0	−7,002
B_1	12°11.5′	0.97745	3.1495	0.21605	0.0796	−0.2009	0.5306	3.6369	0.0573	29.17	−7,855
B_2	23°7.2′	0.91968	2.9633	0.42696	0.2925	−0.3111	0.3912	3.4517	0.2425	65.09	−10,334
B_3	23°7.2′	0.91968	0.7817	0.42696	0.1583	0.3965	1.1740	0.1324	0.2426	65.09	15,544
B_4	11°32.0′	0.97981	0.8328	0.20404	0.0385	0.2004	1.0855	0.0334	0.3417	33.01	16,220
B_5	0	1.00000	0.8500	0	0	0	1.0588	0	0.3750	0	16,459

Table II—Computations for Example 2. Circular Arc Cam With Mushroom Follower

Point	α	ψ	r	$r\omega$	$r\omega^2$	$\sin\psi$	$\cos\psi$	x	Rise	$\dot{x}$	$\ddot{x}$
K_0	115°0′	0	1.8472	174.10	16,408	0	1.00000	0	0	0	16,408
K_1	125°3.4′	10°3.4′	1.8472	174.10	16,408	0.17462	0.98464	0.0284	0.0284	30.55	16,156
K_2	135°6.8′	20°6.8′	1.8472	174.10	16,408	0.34388	0.93901	0.1127	0.1127	59.87	15,407
K_3	135°6.8′	44°53.2′	0.9000	84.82	7,994	0.70571	0.70851	0.2623	0.1127	59.86	5,664
K_4	150°4.5′	29°55.5′	0.9000	84.82	7,994	0.49886	0.86668	0.1200	0.2550	42.31	6,929
K_5	165°2.2′	14°57.8′	0.9000	84.82	7,994	0.25820	0.96609	0.0305	0.3445	21.90	7,723
K_6	180°0′	0	0.9000	84.82	7,994	0	1.00000	0	0.3750	0	7,994

Table III—Computations for Example 3. Straight Sided Cam With Roller Follower

Point	α	θ	r/l	$r\omega$	$r\omega^2$	$\sin\theta$	$\cos\theta$	$r\cos\theta$	$\frac{r}{l}\sin\theta = \sin\phi$	ϕ
B_2	138°10.6′	41°49.4′	0.59019	612.14	5,769.3	0.66684	0.74520	0.48401	0.39356	23°10.6′
B_3	152°7.1′	27°52.9′	0.59019	612.14	5,769.3	0.46764	0.88391	0.57410	0.27600	16°1.2′
B_4	166°3.5′	13°56.5′	0.59019	612.14	5,769.3	0.24094	0.97054	0.63037	0.14220	8°10.6′
B_5	180°0′	0	0.59019	612.14	5,769.3	0	1.00000	0.64950	0	0

Point	$\cos\phi$	$l\cos\phi$	$\tan\phi$	$\sin\theta\tan\phi$	$\cos\theta\tan\phi$	$\frac{m\cos^2\theta}{\cos^3\phi}$	x	Rise	$\dot{x}$	$\ddot{x}$
B_2	0.91929	1.0117	0.42812	0.2855	0.3190	0.4219	0.2543	0.1207	60.35	5,086
B_3	0.96116	1.0578	0.28712	0.1343	0.2538	0.5193	0.1181	0.2569	44.16	7,321
B_4	0.98984	1.0893	0.14369	0.0346	0.1395	0.5732	0.0303	0.3447	23.29	8,707
B_5	1.00000	1.1005	0	0	0	0.5902	0	0.3750	0	9,174

Point	α	α_1	$(r_0 + r_f)\,\omega$	$(r_0 + r_f)\,\omega^2$	$\tan\alpha_1$	$\sec\alpha_1$	$(1 + 2\tan^2\alpha_1) \times \sec\alpha_1$	x = Rise	$\dot{x}$	$\ddot{x}$
B_0	115°0′	0	129.59	12,214	0	1.00000	1.0000	0	0	12,214
B_1	126°35.3′	11°35.3′	129.59	12,214	0.20506	1.02081	1.1067	0.0286	27.13	13,516
B_2	138°10.6′	23°10.6′	129.59	12,214	0.42812	1.08780	1.4866	0.1207	60.35	18,156

dead center, as for contact K_0, θ is equal to 180 deg. and ϕ is equal to zero. Hence Eqs (6) and (7) then become

$$\dot{x} = 0;\quad \ddot{x} = -r\omega^2[1 - (r/l)] \tag{9}$$

Inspection of Figs. 1(B) and (C) shows that the rise L of the follower is

$$L = l\cos\phi + r\cos\theta - (r_o + r_f) \tag{10}$$

From Fig. 1(A) the following relationship is valid

$$r_O + L = r_1 + r_n \tag{11}$$

The cosine theorem applied to triangle OA_1A_2 gives the relationship

$$(r + r_O - r_n)^2 = r^2 + r_1^2 - 2rr_1\cos\beta' \tag{12}$$

In Fig. 1(A) the resisting force of the follower is directed along OB_2. The actuating force of the cam is directed along A_1B_2. This force thus has a sidewise component on the follower, the magnitude of which depends on the value of angle A_1B_2O. This angle is the pressure angle of the cam. The pressure angle must be kept small enough so that the lateral force on the follower is not excessive. Safe values will depend on the speed of operation and resistance of the follower.

EXAMPLE 1. A cam composed of circular arcs is to have a lift L of 0.375 in. for a rotation β of the shaft equal to 65 deg. Speed of the shaft is 900 rpm. Distance between follower center and cam center at maximum lift is to be 1.750 in. Radius r_f of follower roller is 0.600 in., and radius r_n at nose of cam is to be 0.250 in. Find the remaining dimensions of the cam, and compute values for the rise, velocity and acceleration of the follower at different cam positions.

SOLUTION. From a layout of the foregoing data as shown in Fig. 2, the following data can be obtained,

$$\beta' = 180 - 65 = 115 \text{ deg}$$
$$r_O = 1.750 - 0.600 - 0.375 = 0.775 \text{ in.}$$
$$r_O - r_n = 0.775 - 0.250 = 0.525 \text{ in.}$$
$$l_1 = 0.600 + 0.250 = 0.850 \text{ in.}$$
$$r_1 = 1.750 - 0.850 = 0.900 \text{ in.}$$

Substituting these values in Eq (12)

$$(r + 0.525)^2 = r^2 + 0.9^2 - 2r \times 0.9 \cos 115 \text{ deg}$$

$$r = 1.8472 \text{ in.}$$

The remaining computations are given in Table I. Roller position at B_1 is located on the bisector of angle B_1OB_2. Roller position at B_3 is located on the bisector of angle B_2OB_4. Two sets of computations can be made for B_2, one for each of the two slider cranks for this position. The calculations should give identical values for the rise and velocity. There will be two values, however, for the accelerations.

The values for rise, velocity, and acceleration are plotted in Fig. 3. The equations give results with positive and negative signs consistent with the action of a slider crank. These have been disregarded in plotting Fig. 3, an upward motion of the follower has been taken as positive and downward motion as negative.

Circular Arc Disk Cams With Mushroom Follower

In Fig. 4 is shown a disk cam composed of circular arcs actuating a mushroom follower. Arc K_0K_2 has its center at A_1 and arc K_2K_4 has its center at A_2. Distance A_1K_1 in Fig. 4(A) is constant for all points of contact between K_0 and K_2. Hence the vertical movement of the follower is identical with the vertical movement of center A_1. A Scotch yoke located at A_1 would therefore have the same motion as the follower. Where OA_1 equals r_1 for contact at K_1 the follower has risen from its lowest position an amount x equal to

$$x = r_1(1 - \cos \psi) \quad (13)$$

Successive differentiations give the equations for velocity and acceleration:

$$\dot{x} = r\,\omega \sin \psi \quad (14)$$
$$\ddot{x} = r\,\omega^2 \cos \psi \quad (15)$$

In a similar manner, for arc K_2K_4, the follower has the same motion as a Scotch yoke attached at A_2. Where OA_2 equals r_1, for contact at K_3, Fig. 4(B), the follower has dropped from its highest position an amount x_1 equal to

$$x_1 = r_1(1 - \cos \psi_1) \quad (16)$$

Successive differentiations give the equations for velocity and acceleration:

$$\dot{x}_1 = r_1\,\omega \sin \psi_1 \quad (17)$$
$$\ddot{x}_1 = r_1\,\omega^2 \cos \psi_1 \quad (18)$$

This type of cam as shown by Eqs (17) and (18) has simple harmonic motion.

EXAMPLE 2. Compute and plot the values for rise, velocity and acceleration of a mushroom follower actuated by the cam of Example 1.

SOLUTION. This cam is shown in Fig. 5. Angle $K_0A_1K_2$ is divided into two equal parts, and angle $K_2A_2K_4$ is divided into three equal parts. The computations are given in Table II. To compare the action of a mushroom follower with that of a roller follower when both are moved by the same cam, the rise, velocity, and acceleration are plotted in Fig. 3 along with the results of Example 1. The results from Eqs

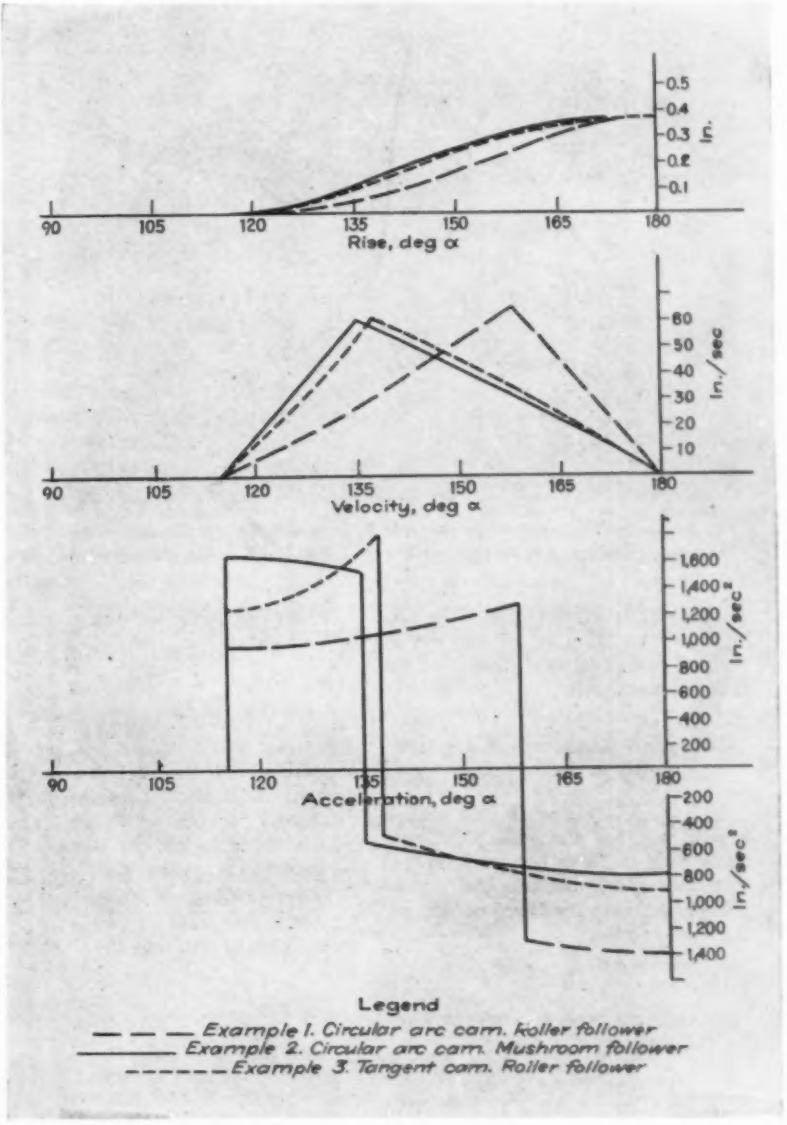


Fig. 3—Plots of the rise, the velocity and the acceleration for three types of cams.

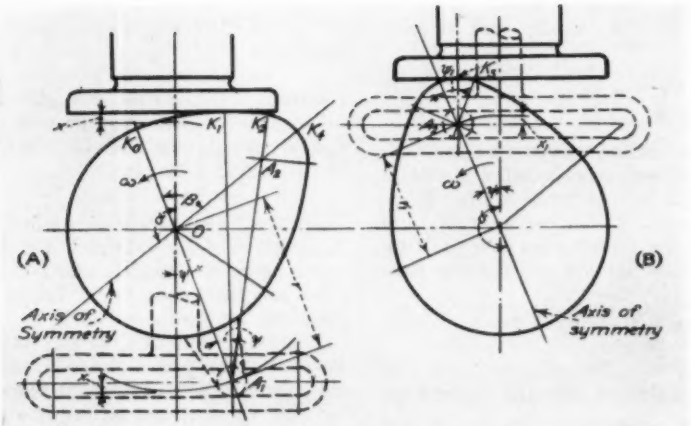


Fig. 4—(A) Circular arc cam and equivalent Scotch yoke when follower is on arc K_1K_2. (B) Equivalent Scotch yoke linkage when follower is on nose.

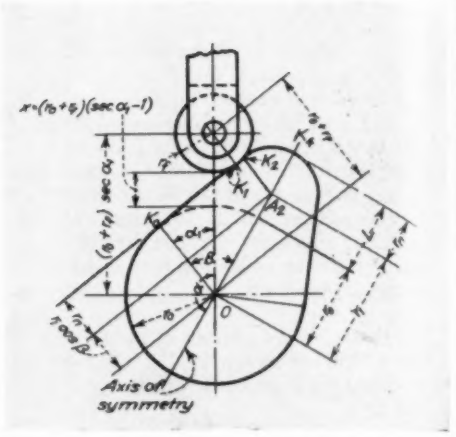


Fig. 6—Cam composed of straight or tangential sides and circular arcs with roller follower.

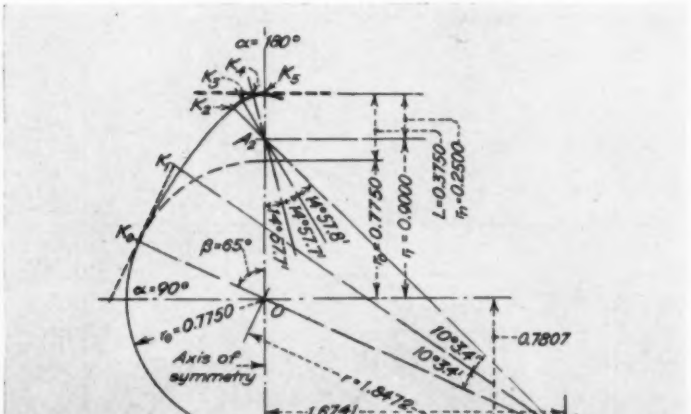


Fig. 5—Layout for solution of Example 2, in which the circular arc cam of Example 1 actuates a mushroom follower. Table II gives the computations.

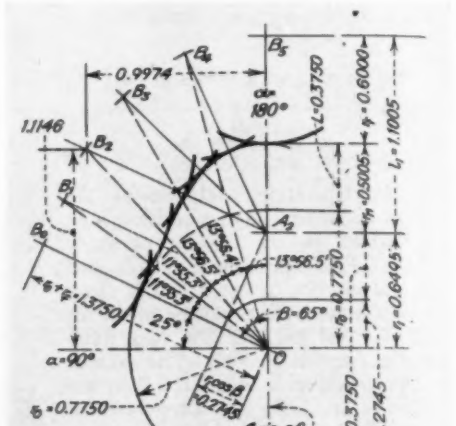


Fig. 7—Layout for the solution of Example 3. Cam is straight sided and has a roller follower.

(13) to (18) are interpreted by taking upward motion of the follower as positive and downward motion of the follower as negative.

Cam With Tangential Sides and Roller Follower

In Fig. 6 is shown a cam with a straight side for K_0K_2. Arc K_2K_4 is a circle with center at A_2; previous results apply for contact in this region.

Additional equations are needed for the K_0K_2 portion.

$$x = (r_O + r_f)\sec\alpha_1 - (r_O + r_f) = (r_O + r_f)(\sec\alpha_1 - 1) \quad (19)$$

Differentiation gives the velocity $\dot{x}$ for the follower,

$$\dot{x} = (r_O + r_f)\,\omega \sec\alpha_1 \tan\alpha_1 \quad (20)$$

Differentiation gives the acceleration $\ddot{x}$,

$$\ddot{x} = (r_O + r_f)\,\omega^2(1 + 2\tan^2\alpha_1)\sec\alpha_1 \quad (21)$$

From Fig. 6 the following relationships are apparent,

$$r_O - r_n = r_1 - L = r_1\cos\beta \quad (22)$$

$$L = r_1(1 - \cos\beta) \quad (23)$$

EXAMPLE 3. Compute and plot the values for rise, velocity, and acceleration of the follower actuated by the tangential cam shown in Fig. 7, for which the data are the same as for Example 1 except that a suitable value for radius r_n must be found.

SOLUTION: From Eq(23), radius r_1 is

$$r_1 = \frac{L}{1 - \cos\beta} = \frac{0.375}{1 - 0.42262} = 0.6495 \text{ in.}$$

$l_1 = 1.75 - 0.6495 = 1.1005$ in.

$r_n = l_1 - r_f = 1.1005 - 0.6000 = 0.5005$ in.

$r_O = r_n + r_1\cos\beta = 0.5005 + 0.6495 \times 0.42262 = 0.7750$ in.

The remaining computations are given in Table III and the results plotted in Fig. 3. As before, upward motion of the follower is positive, and downward motion is considered negative.

Any of the cams of the three foregoing examples might be used in a given application. The curves of Fig. 3 permit a comparison of the performance of the three types. The circular arc cam with mushroom follower and the tangent cam with roller follower raise their followers more quickly than the circular arc cam with roller follower. The circular arc cam with roller follower has a slightly greater change in acceleration when the follower makes contact at the junction between the arc of different curvatures, consequently, the inertia forces in the mechanism will be slightly larger with this type of cam.

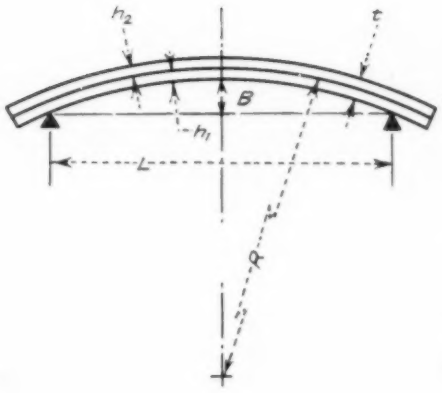


Fig. 1—Curvature of arc can be readily calculated for straight, narrow strip of thermostatic bimetal.

$$\frac{1}{R} = \frac{6\,(\alpha_2-\alpha_1)\,(\Delta T)\,(1+m)^2}{t\left[3(1+m)^2+(1+mn)\left(m^2+\frac{1}{mn}\right)\right]}$$

where:

R = radius of curvature

α_1, α_2 are the lower and higher coefficients of linear expansion, in./in./F

ΔT is the temperature change, F

$m = \frac{h_1}{h_2}$, ratio of metal thickness

t = overall thickness of bimetal strip, inches

$n = \frac{E_1}{E_2}$, ratio of moduli of elasticity

Designing

Bimetal Control Devices

With many more combinations of bimetals now available, more efficient and economical design of equipment can be made. Applications, design formulas and tabular data for various types of temperature actuated devices.

U. SAVOLAINEN
Engineer, General Plate Div.,
Metals & Controls Corporation

IN DESIGNING A THERMOSTATIC ELEMENT for a thermometer, thermostat or other thermally operated device, the problem is one of choosing the most economical type of material and most effective shape of element. In addition, this element must have the required thermal activity—movement in response to temperature change—or a combination of both movement and force. The material having the highest thermal activity in the working temperature range is usually chosen, although this is not always true. If a thermostat is required to operate over a wide temperature range, a low thermal activity material may be best. Where the length of element is specified for a given temperature range, a thin, low thermal activity material may be required. Increasing the bimetal thickness may be most effective in reducing the movement of a high thermal activity element. Besides thermal activity, factors such as corrosion resistance, thermal conductivity and electrical resistance influence the choice of material and design of the element.

A bimetallic element consists of a laminated strip of two alloys having different coefficients of thermal expansion. A temperature rise causes the alloy with the higher coefficient of expansion (called expansitivity) to expand more than the other, but since these two alloys are bonded together such linear expansion is restrained. This restraint of one member against the other causes internal stresses which produce a curvature of the entire element. If the temperature is uniform throughout the strip, its curvature will be a circular arc.

For a given temperature change, the curvature of an element is dependent upon several factors; they are: difference in expansitivities, $(\alpha_2 - \alpha_1)$; ratio of the alloy thickness, h_1/h_2; and elasticity ratio of the alloys, E_1/E_2. Fig. 1 illustrates this and gives the formula for calculating the radius of curvature. If the metal thickness be equal ($h_1 = h_2$) and the elasticities the same ($E_1 = E_2$), then $m = 1$ and $n = 1$ and this curvature formula reduces to:

$$\frac{1}{R} = \frac{3}{2}\,\frac{(\alpha_2 - \alpha_1)\,\Delta T}{t}$$

This formula illustrates that the curvature is directly proportional to the difference in coefficients of linear expansion and to the change in temperature. The curvature is inversely proportional to the overall thickness of the bimetal strip. This simplified formula is usually applicable because most bimetal thermostats are made of equal thickness elements. Using an elasticity

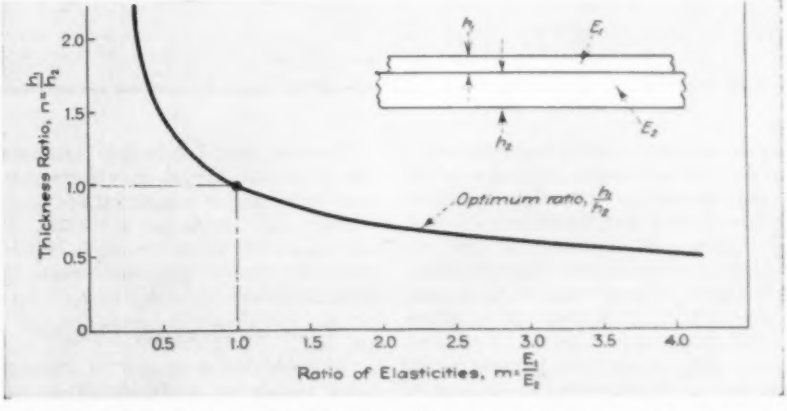
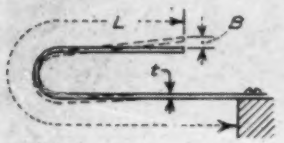
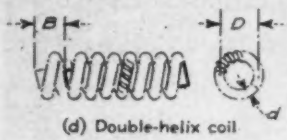
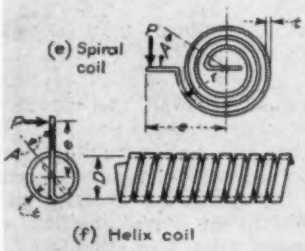


Fig. 2—Thickness of bimetallic element should be altered when elasticities are greatly different. Recommended thickness change is illustrated for different elasticity ratios.

Table I — Formulas to Determine Force, Deflection, and Stress of Thermostatic Bimetal Element

For specific value of constants see Table III	Unrestrained Deflection	Restrained Force	Ratio $\frac{P}{B}$	Stress
LINEAR MOTION				
(a) Cantilever strip	$B = \frac{c_2 (T_2 - T_1) L^2}{t}$	$p = \frac{b_2 (T_2 - T_1) wt^2}{L}$	$\frac{P}{B} = \frac{a_2\ wt^3}{L^3}$	$S = \frac{6PL}{wt^2}$
 (b) U-shape	$B = \frac{c_2 (T_2 - T_1) L^2}{2t}$	$p = \frac{2b_2 (T_2 - T_1) wt^2}{L}$	$\frac{P}{B} = \frac{4a_2\ wt^3}{L^3}$	$S = \frac{3PL}{wt^2}$
(c) Simple beam	$B = \frac{c_2 (T_2 - T_1) L^2}{4t}$	$p = \frac{4b_2 (T_2 - T_1) wt^2}{L}$	$\frac{P}{B} = \frac{16a_2 wt^3}{L^3}$	$S = \frac{3PL}{2wt^2}$
 (d) Double-helix coil	$B = \frac{c_3 (T_2 - T_1) LD}{t}$	$p = \frac{b_3 (T_2 - T_1) wt^2}{D}$	$\frac{P}{B} = \frac{a_3\ wt^3}{LD^2}$	$S = \frac{3PD\left(D + \frac{d}{2}\right)}{wt^3}$
ANGULAR MOTION				
 (e) Spiral coil (f) Helix coil	$A = \frac{c_1 (T_2 - T_1) L}{t}$	$p = \frac{b_1 (T_2 - T_1) wt^2}{r}$	$\frac{P}{A} = \frac{a_1\ wt^3}{Lr}$	$S = \frac{6P(r+e)}{wt^2}$

where:

- t — Thickness, in.
- w — width, in.
- L — Active length of strip, in.
- a_1 — Torque constant, coils
- a_2 — Force constant, strip, & U-shape
- a_3 — Force constant, double-helix coil
- b_1 — Torque-temperature constant, coils
- b_2 — Force-temperature constant, strip, & U-shape
- b_3 — Force-temperature constant, double helix coil
- c_1 — Thermal deflection constant, coils
- c_2 — Thermal deflection constant, strip, & U-shape
- c_3 — Thermal deflection constant, double-helix coil
- B — Deflection, strip, in.
- A — Angular rotation, coils, deg
- P — Force, lb
- e — Radius at point load is applied, in.
- $(T_2 - T_1)$ — Temperature change, F
- s — Stress, psi
- F — Flexitivity, ASTM value 50-200 F
- r — Radius of outside coil, in.

ratio of $m = 1$ gives maximum curvature and any slight variations in this ratio do not greatly affect the curvature of the bimetallic element.

When the elasticities of the two alloys are appreciably different, a compensating change in the thickness ratio, m can be made for best performance. The optimum thickness ratio for various ratios of elasticity is given by the curve in Fig. 2. The importance of this graph, can be illustrated by a case where $E_1/E_2 = 4.0$, altering the thickness ratio from 1.0 to 0.35 increased by 27 percent the work performed by the bimetal for a given temperature change. The work that a bimetal can do is proportional to the deflection constant *squared* times the torque or force constant.

Types of Bimetallic Elements

Many different shapes of thermostatic metals are obtainable. They can be divided into six basic types as shown in Table I, which presents the

Table II—Allowable Working Stresses for Ferrous Alloy Elements Operating at Different Temperatures

Max operating Temperature F.	Allowable Working Stress in psi
75	25,000
300	23,000
500	18,000
700	13,000
900	5,000
1000	0 to 1,000

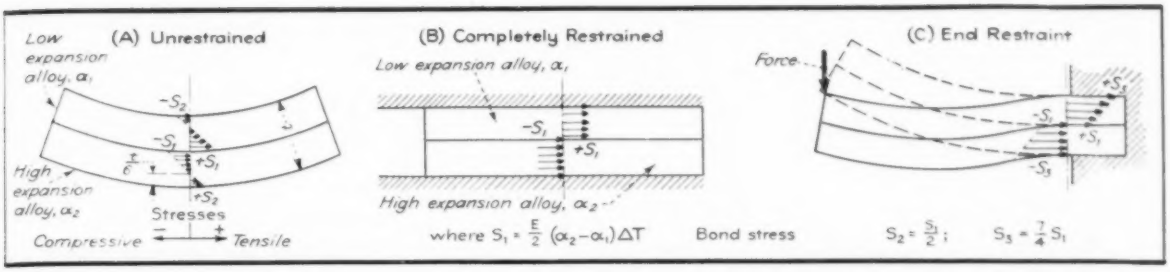


Fig. 3—Internal stress distribution in bimetallic strips is affected by a temperature rise, difference in expansitivities, and constraint: (A) unrestrained strip has arc of uniform curvature; (B) completely restrained; (C) cantilever exerts force due to end-restraint.

Table III—Thermostatic Properties of Most Common Bimetals

Type	CONSTANTS FOR EQUATIONS IN TABLE I									FLEXIVITY F	ELECTRICAL RESISTANCE r	ELASTICITY E	REMARKS
	a_1	a_2	a_3	b_1	b_2	b_3	c_1	c_2	c_3				
	x 10^3	x 10^6	x 10^8				x 10^{-5}	x 10^{-7}	x 10^{-7}	x 10^{-7}	ohms/cir mil ft	x 10^6	
A1	27.5	4.50	5.90	29.0	37.5	43.5	105	83.0	74.0	156	90	18.0	High thermal deflection rate but maximum operating temperature is limited to 350 F.
B1	34.5	5.70	7.40	33.5	44.5	50.5	97.5	78.0	68.0	145	465	22.8	High deflection rate—general purpose type that is widely used having a high strength. High temp use to 1000 F.
B3	35.0	5.75	7.50	25.0	33.0	38.5	72.0	57.5	51.5	108	410	23.0	Moderate rate of thermal deflection
E1	34.5	5.70	7.40	31.0	40.5	46.0	90.0	71.0	62.5	135	500	22.8	High rate of deflection, low temp range, —100 to 300 F.
E5	38.0	6.20	8.15	15.0	19.0	22.0	40.0	31.0	27.5	60	345	24.8	Lowest deflection rate, high temp range to 1000 F.; used in toasters and irons. Also used in thermometers.
G7	40.0	6.55	8.60	16.5	21.5	25.0	41.5	33.0	29.5	62	465	26.2	Corrosion resistant to steam, water, moisture. Can be used to 1000 F. Low rate of thermal activity.
J1	29.0	4.75	6.20	26.0	33.5	3.90	90.0	71.0	63.0	135	135	19.0	General purpose, medium cost type. Max temp about 625 F. Used on automotive automatic chokes.
J7	34.0	5.50	7.30	13.0	16.5	19.5	38.0	30.0	26.5	57	130	22.0	Corrosion resistant type similar to G7. Used in water and steam mixing valves.
N1	37.5	6.15	8.05	26.0	33.5	3.90	69.0	54.5	48.5	103	102	24.6	High temperature use to 1000 F. For high electrical rating devices such as circuit breakers.
F20R	30.0	4.90	6.45	27.0	35.0	40.5	90.0	71.0	63.0	134	20	19.6	Low electrical resistance series used for circuit breakers and line protectors with relatively high current.
F100R	30.0	4.90	6.45	27.0	35.0	40.5	90.0	71.0	63.0	134	100	19.6	
B125R	34.5	5.70	7.40	27.5	36.0	41.5	80.0	63.0	56.0	120	125	22.8	Intermediate and higher range of electrical resistance. Similar applications as F series.
B400R	34.5	5.70	7.40	33.5	44.0	50.0	97.5	77.5	68.0	145	400	22.8	
P675R	27.0	4.50	5.85	36.5	48.5	56.0	135	108	96.0	203	675	18.0	Highest rate of thermal deflection with high electrical resistance.
P850R	27.0	4.50	5.85	27.0	36.0	41.5	100	80.0	71.0	150	850	18.0	Highest electrical resistance.
6010	35.0	5.75	7.50	21.5	28.0	32.5	62	49.0	43.5	93	208	23.0	Special purpose material; thermal deflection rate reverses at about 500 F.

Note: Constants b and c are based on a temperature range of 50 to 200 F., while values for a and r are for 70 F.

elements in two groups depending upon the type of motion produced: linear or angular. The straight strips (a), (b) and (c) give essentially linear motion for small temperature changes. The spiral and helix coils (e) and (f) produce rotation and are therefore particularly adapted for thermometers or dial-type devices. The spiral coil is the most compact type of element.

The double helical coil (d) either lengthens or shortens with temperature increase, depending upon the direction of primary and secondary coiling, also whether the higher expansion element is inside or outside. Disregarding a slight angular rotation, this device will produce true linear motion. Because of the long length of strip metal making up the coil, it has a high rate thermal activity and relatively high electrical resistance. For linear motion it is the most compact design for a given length of bimetal.

Formulas are given in Table I to calculate the deflection and force for each type of thermostatic element. Cantilever types (a) and (b) have the simplest construction for linear motion and the spiral coil (e) is the simplest for angular motion. For a given length and cross sectional area, the cantilever-type element (a) gives twice the motion of a simple beam and 4 times that of the U-shape. However, it is to be noted that the force relationship is inversely proportional to the displacement. The total work produced

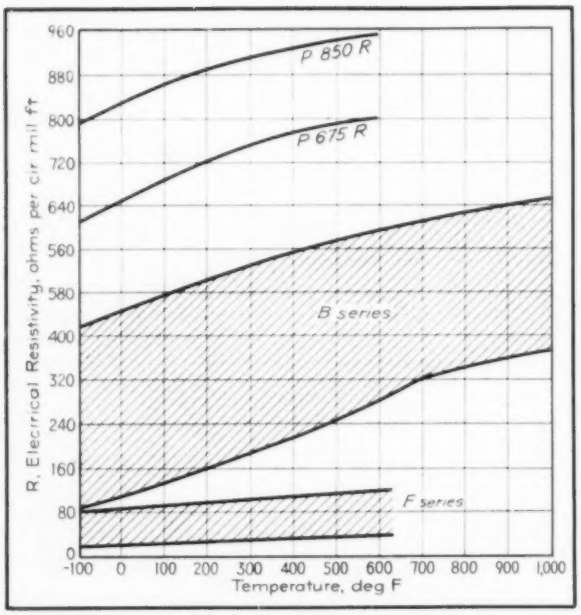


Fig. 4—Choice of bimetal for electrical applications is simplified by a large selection of electrical resistances which can be used over a wide operating temperature range up to 1000 F.

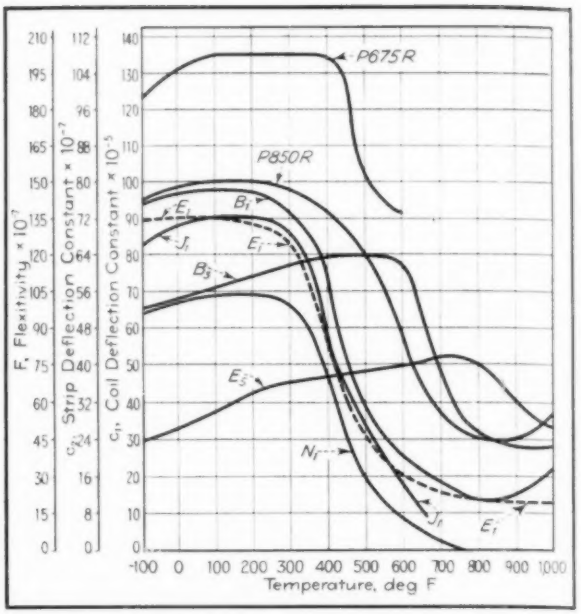


Fig. 5—Choice of bimetal is based on the thermal activity under operating temperature conditions. The instantaneous thermal activity constants for typical bimetals are illustrated.

by each of the first three types would be identical. For limited space applications the double helix coil has greater possibilities since a much greater length of bimetal is possible, but it is more complex in shape. Of the angular types (e) and (f), the choice of either would be dictated by the space limitation since the formulas for both are identical.

Internal Stresses

The curved shape of a heated bimetallic element is a result of the internal stresses which are created by the thermal expansion of one component that is restrained by the other. In addition, there may be stresses that are produced by mechanical loads. To insure reliable and durable performance both of these stresses must be considered. Residual stresses which are caused by the manufacturing processes, such as rolling, bonding, flattening, and heat treating, cannot be calculated and therefore must be compensated for in terms of a lower allowable stress.

Internal thermal stresses created in a simple bimetal beam are shown in Fig. 3. If this strip is uniformly heated and not restrained, it will become deformed to the extent given by the movement formula, see Table I (c). The magnitude and distribution of these internal tresses are shown in Fig. 3 (A). If this strip were fully restrained as in Fig. 3 (B), the internal stresses would change as shown. In a cantilever thermostat which is mechanically restrained at the free end, the internal stress would be a maximum near the support; see Fig. 3 (C). Note that the magnitude of the stress at the bonding surface is identical in each case.

Cantilever strip of uniform cross section is not efficient since it is worked to full capacity only at the clamped end. A tapered cantilever of uniform strength will give, for the same volume of thermostat metal, 33 percent greater contact force and 43 percent less stress. This type gives better economy due to uniform stress distribution throughout the length. Another desirable feature of tapered strips is their tendency to lessen troublesome vibrations as a result of the mass being concentrated nearer support.

Stresses caused by mechanical loading can be calculated considering the element as a single or isotropic material and using the usual formulas. This is not theoretically correct but serves for all practical purposes. The modulii for different bimetals listed in Table III has been experimentally determined and can be used when necessary. Since the stress that a thermostat metal element can stand without taking a permanent set depends upon the temperature and previous history such as the rolling, forming and heat treatment, high safety factors must be used to arrive at the allowable stresses; values are given in Table II.

Force vs. Deflection

There are three general ways which thermostatic elements are used: (1) Free deflection with no restraint; (2) Force exerted while completely restrained; (3) Combination of (1) and (2) where both the force and deflection are specified. The formulas in Table I can be used to calculate items (1) and (2); item (3) requires further explanation.

Since the temperature change is responsible for either deflection or force, it can be used to solve the problem where both deflection and force are involved. This can be done by considering that a part of the temperature change causes the required deflection and the remainder develops the pressure. Any part of the temperature change may be applied either way but the size of the thermostat will depend upon the proportions used. However, for best design and for maximum effectiveness, one half of the temperature change should be used to produce the deflection and the other half for developing the force.

Electrical Thermostats

An increasing number of products are made, such as thermal relays, circuit breakers, and motor overload protectors in which heat generated by passage of an electric current through a thermostat metal element operates the device. Such applications require a variety of thermostat metals covering a wide range of electrical resistivities. These devices were formerly all made with the same type of bimetal, widely different ratings being obtained by varying the size and shape of the element. But the manufacturers de-

sired to make a line of circuit breakers all of the same physical size but by changing the element material. A line of thermostatic metals was developed to control the power in uniform steps from low to high.

The heating effects of an electric current is:

$$H = \frac{ri^2t}{4.18}$$

where H is in calories, r is Resistance in ohms, i is current in amperes, and t is time in seconds.

The temperature rise (disregarding heat losses) of a resistor is given by:

$$T = \frac{0.43\ ri^2t}{MS}$$

where T is the temperature rise in deg F, M is the mass in grams, and S is the specific heat.

Note that the heating effect is proportional to i^2 in which i^2 is analogous to the current rating of a breaker and r is analogous to the electrical resistivity of the thermostat metal. Throughout a line of circuit breakers, the heating effect which is proportional to i^2 must be a constant for uniform tripping time. Therefore, for a given bi-metal element the series of thermostat metals must vary in electrical resistivity as the square of the rating of the circuit breaker. The following equation is useful in designing a line of breakers since after experimentally determining the type of material for one rating the others can be approximately calculated by means of:

$$I^2R = \text{Constant}$$

where I is the Current rating of breaker and R is the Resistivity of thermostat metal.

This equation is useful also in determining whether the design of the line of breakers is such as to make best use of the range of resistivity materials available. The highest rating should use the lowest resistance material and the lowest rating the highest resistance material.

The range of electrical resistivities available before the advent of special electrical restivity series is illustrated by the thermostat metals listed in the following table:

Thermostat Metal	Electrical Resistivity at 75 F. in Ohms per cir mil ft
W	90
X	95
Y	345
Z	500

The special thermostat metals developed to meet these needs were based on standard thermostat metals such as type *Z* in above table. The resistivity was varied by incorporating a third layer of variable thickness of a relatively low resistivity metal. This filled in the range from 125 to 470 ohms cir mil ft. A series from 20 to 100 ohms per cmf. utilized high conductivity copper alloys for the third layer. The resistivity of such an element can be calculated easily considering the three laminations as a parallel circuit:

$$\frac{x}{r_1} + \frac{y}{r_2} + \frac{z}{r_3} = \frac{100}{R}$$

where: x, y and z are the thicknesses of components in percent of the total thickness of the three; r_1, r_2 and r_3 are the resistivities of the three components; and R is the resistivity in ohms per cir mil ft of the combination.

Fig. 4 gives the resistivity vs. temperature of a series of electrical thermostatic metals whose resistivities at 75 F cover the range from 20 to 850 ohms/cir mil ft. The resistance vs. temperature information is important in design work since it provides: (1) the resistance at the operating temperature, (2) the rate of increase with temperature rise (coefficient of resistance) and (3) the resistance at room temperature.

Important Properties of Alloys

In chosing the metal components of the thermostat, several factors must be kept in mind to produce an efficient, economical, and accurate finished product. First, since the rate of thermal activity is proportional to difference in the coefficient of expansion (called expansitivity) of the two components, materials must be chosen to have as low and as high expansitivities as possible. Also the rate of change of the expansitivities must be such to give the required linearity, or required deviation from linearity of thermal activity over the required operating temperature range. Usually a bimetal with a constant expansitivity is desirable; although for special applications varying expansitiviites are used to give the required thermal activity.

A list of typical bimetallic materials is given in Table III; included are the data essential to solve the force and deflection equations given in Table I. Since the values for flexitivity and the constants in this chart apply to a temperature range of from 50 to 200 F, additional information is required to design for a higher operating range. For this range, Fig. 5 should be used.

Measuring Thermal Activity

The thermal activity or "flexitivity" as it is called in ASTM standards, is one of the most important factors of bimetallic elements and is defined as, "the change in curvature of the longitudinal center line of the specimen per unit temperature change per unit thickness." It is given by the formula in Fig. 6. If the bimetallic element is initially straight, or $1/R_1 = 0$, this formula reduces to

$$\text{Flexitivity, } F = \frac{\left(\frac{1}{R}\right)t}{\Delta T}$$

Today it is possible to obtain bimetals which possess almost any characteristic desired. Such special data can be obtained from the manufacturer of bimetals. However, data in Table III covers the most important alloys.

After choosing the type of material, the shape of element has to be determined such that it will move in a proper manner to perform the required operation and that it will fit into the space available.

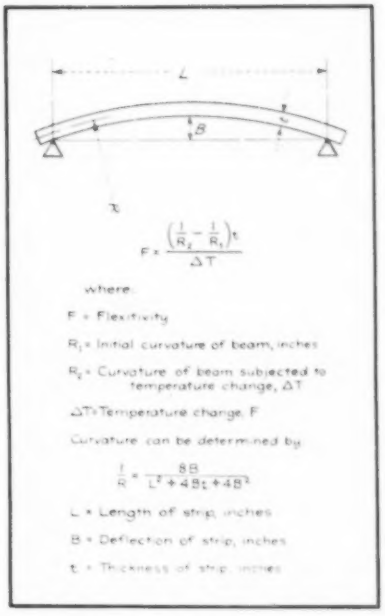

L
B
t
x
$F = \frac{\left(\frac{1}{R_2} - \frac{1}{R_1}\right)t}{\Delta T}$
where
F = Flexitivity
R_1 = Initial curvature of beam, inches
R_2 = Curvature of beam subjected to temperature change, ΔT
ΔT = Temperature change, F
Curvature can be determined by
$\frac{1}{R} = \frac{8B}{L^2 + 4Bt + 4B^2}$
L = Length of strip, inches
B = Deflection of strip, inches
t = Thickness of strip, inches

Fig. 6—Flexitivity is found by measuring deflection of bimetal strip which has been subjected to a change in temperature.

Manufacturing Tolerances

In calculating the proportion and sizes of bimetallic elements almost any thickness can be made. To facilitate the dimensioning, Table IV was prepared which gives standard manufacturing tolerances of this material.

Table IV—Standard Manufacturing Tolerances of Bimetallic Materials

Thickness, in.	Tolerance, ± in.	Width, in.	Tolerance, ± in.
0.005 to 0.010	0.00035	up to ½	0.004
0.010 to 0.015	0.00040	½ to 1	0.005
0.015 to 0.020	0.0005	1 to 3	0.006
0.020 and up	2½%	3 to 5	0.010
		5 to 8	0.015

Control Mechanisms for

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Franklin Institute Laboratories for Research and Development

RADIAL PUMPS

HYDRAULIC APPLICATIONS have increased with improved flow controls. The most effective method of controlling the fluid is at the pump. The design of both the radial and axial piston pumps lend themselves to variable-delivery by simply changing the pump displacement. Thus, the output can be altered without changing the speed of the motor. This method eliminates the power loss that is caused by either throttling or bypassing fluid. In addition, such a control device can change the direction of flow without requiring special valves or altering the *direction* of the driving motor. With these controls, the pump is well adapted for both heavy-duty service or for accurate drive control of such equipment as machine tools. Sketches illustrate the method by which the pump displacement and pressure are regulated.

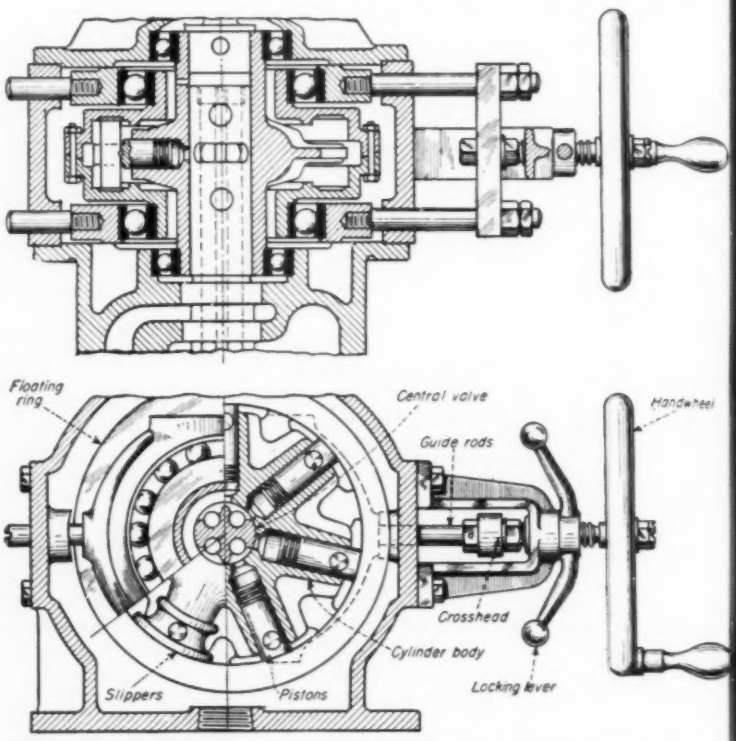


FIG. 1—MANUAL DISPLACEMENT CONTROL by handwheel screw move floating ring to an off-center position. Cylinder body is driven on-center and an eccentricity of the floating ring causes relative motion between the pistons and cylinders during each revolution. Adjusting floating ring changes flow from zero to maximum in either direction. This type is used where high pressures are required.

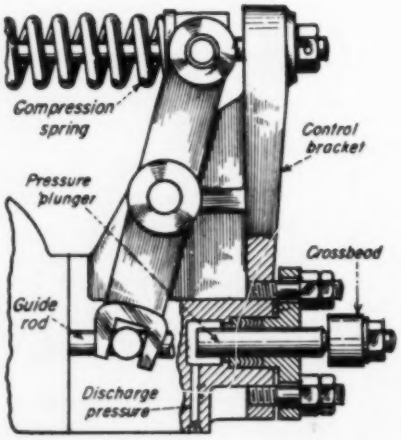


FIG. 2—AUTOMATIC PRESSURE REGULATION is performed by balanced forces acting between plunger and compression spring. Pump remains in full stroke position until pressure on plunger overcomes the spring force and returns pump to a leakage stroke which sustains the pressure. Adjustable nut on spring sets discharge pressure.

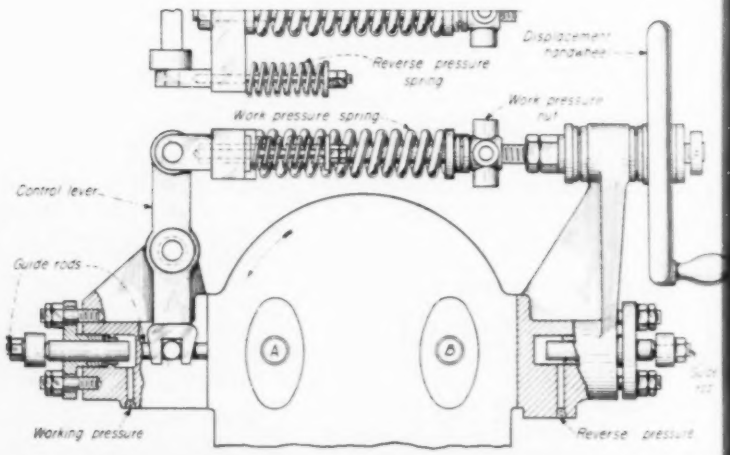
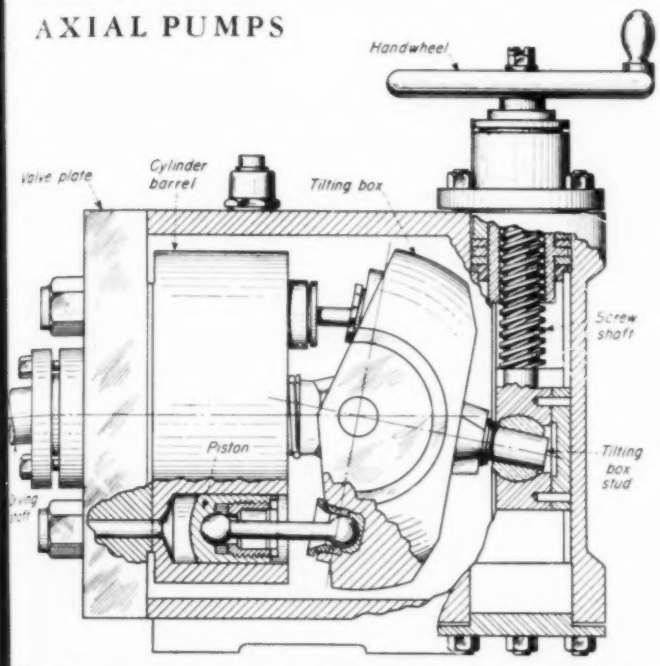


FIG. 3—REVERSING CONTROL is combined with displacement and pressure d vice. Clockwise handwheel rotation increases flow from flange *A*. Work-spring for determines discharge pressure. Counter-clockwise handwheel motion produc flow from flange *B*; thus reversing flow. Device permits wide range of worki pressure that is independent of pull-back pressure, which is usually constant.

Hydraulic Piston Pumps

AXIAL PUMPS



...4—MANUAL DISPLACEMENT CONTROL consists of handwheel and ...w shaft which alters angle of tilting box, or "wobble plate." Larger ...le increases the piston stroke and produces higher flow. Reversed flow ...ssible by simply changing angle to other quadrant. Axial piston pumps ...often used where high displacements at lower pressures are necessary.

Data by courtesy of American Engineering Co., Philadelphia, Pa.

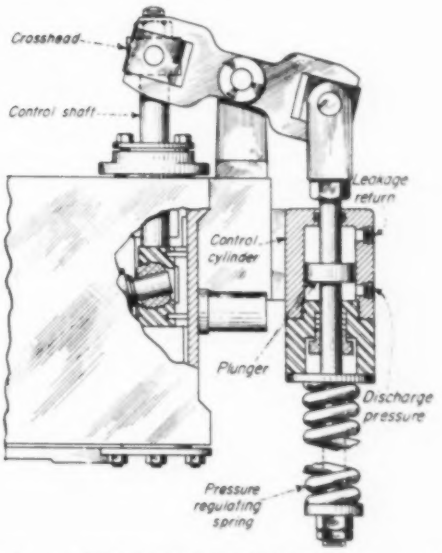
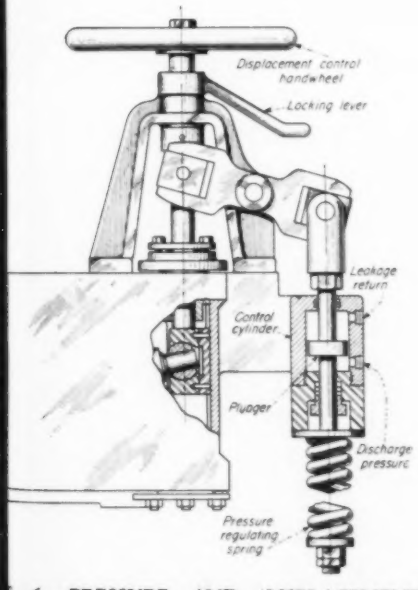


FIG. 5—DISCHARGE PRESSURE is automatically controlled by balanced forces acting between pressure plunger and adjustable compression spring. Pump remains in full stroke position until pressure on plunger overcomes spring force and returns the tilting box stud toward the neutral position.



...6—PRESSURE AND DISPLACEMENT ...NTROL is obtained by combining mechanism in ...5 with a handwheel. Each control operates in...ndently. Handwheel regulates displacement ...piston-spring mechanism maintains pressure.

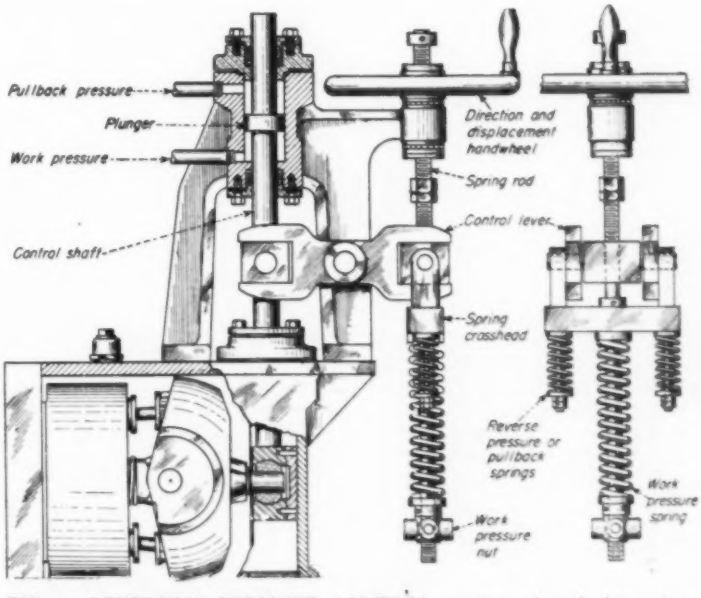


FIG. 7—REVERSING PRESSURE CONTROL consists of a double-acting plunger and springs in addition to the device in Fig. 6. Clockwise motion of handwheel moves tilting box stud below center, and causes pump to deliver fluid at working pressure which is controlled by work spring. Counterclockwise wheel motion moves box stud down and produces reverse flow.

Design and Construction Of Wide Face Steel Pulleys

A design analysis of low cost wide face steel pulleys. Types of hub constructions. How general proportions and operating conditions affect rim and disk thickness. The author herein presents data not previously available— and data that is necessary for proper design to prevent arm breakage and "walking" on the shaft.

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ONE OF THE SIMPLEST TYPES of power transmission elements, the pulley, has taken on increased importance with the advent and widespread acceptance of industrial belt conveyor systems. When designing such pulleys the problem is to provide sufficient strength in the rims and disks to prevent breakage due to high stress, and at the same time provide for sufficient flexure in both the shaft and pulley that neither breaks due to shock loading, or that the pulley "walks" on the shaft.

Hub Constructions

Up until about 20 years ago, almost all conveyor pulleys were made of cast iron and were designed with a two-arm construction, Fig. 1(A). Almost all manufacturers of these pulleys had some difficulty with arms breaking, and many of them discovered that the cause of this breakage was due to the tendency of the shaft to deflect and carry the hubs with it.

Later, interchangeable-hub welded-steel conveyor pulleys, Fig. 1(B), appeared on the scene. These pulleys use interchangeable bolted hubs that have tapered shoulders accurately mating with similar tapered recesses in the disks. Hubs are held in position by six or eight bolts, depending upon the hub size. The reason for the tapered fit is to provide a positive engagement between the hub and the disk to withstand the direct belt-pull load which must be transmitted from the rim of the pulley through the disk and hub onto the shaft. The advantage of replaceable construction is that stock pulleys can be adapted to the shaft size that is made necessary by loading and bearing spacing and custom design-manufacture is not required.

When pulleys are made to order for specific application it is almost as easy and the cost is about the same to employ a construction as shown in Fig. 1(C) in which the hubs are solid steel and are completely welded into the end disk. This construction has been found to be most satisfactory for heavy duty and unusual applications. Also, accurate calculation of the disk stresses is relatively easy with this type of unit, whereas the bolted-hub design leaves something to be desired in this respect.

Disk Stresses

Perhaps the initial attack point in designing a welded steel pulley is the pulley disk. A steel pulley with a thin disk, Fig. 2(A), allows the shaft to deflect as it naturally would by virtue of the two loads applied to it through the hubs. The disk is thin enough that it does not restrain the shaft appreciably. A pulley made with thick disks that do not bend appreciably as a result of the bending moments applied to them causes another type of shaft deflection—from the hub outward—and for all practical purposes the shaft as shown in Fig. 2(B) could just as well be cut off between the two hubs. Between these two extremes lies a range of thicknesses that cause high disk stresses, unnecessarily high bending moments on the hub, and eventually pulley breakage.

Table I—Disk Constants for Various Disk-to-Hub Diameter Ratios

Ratio of Hub Dia to Disk Dia d/D	Disk Constants	
	β	α
0.1	6.06	0.87
0.15	5.40	1.23
0.20	4.52	1.68
0.25	3.40	2.31
0.30	2.65	3.10
0.35	2.15	4.00
0.40	1.75	5.45
0.45	1.45	8.20
0.50	1.14	12.40
0.55	0.888	18.00
0.60	0.684	28.5
0.65	0.568	44.0
0.70	0.466	77.9
0.75	0.346	156.
0.80	0.262	314.

Using basic data obtained from Roark's "Formulas for Stress and Strain," Second Edition, page 210, the following formula for the disk stress has been developed:

$$\delta_r = \frac{(P)\,(\beta)\,(L - X)\,(X)}{(4)\,(D)\,(\beta)\,(I_s)\left(\frac{1}{\alpha t^3} + \frac{X}{2I_s}\right)} \qquad (1)$$

where δ is the radial stress in the disk at the hub, lb/in.^{2}; P is the total resultant belt pull on the pulley, lb; d is the hub diameter, in.; D is the disk diameter, in.; α is the disk constant, Table I; β is the disk constant, Table I; L is the center-to-center distance between bearings, in.; X is the center-to-center distance between disks, in.; t is the disk thickness, in.; and I_s is the moment of inertia of shaft, inches^4.

To find the thickness that produces the maximum stress, the previous formula is differentiated with respect to disk thickness, t, and then set to zero, thus yielding the point on the curve having maximum stress. This relationship is:

$$t^3 = \frac{I_s}{\alpha X} \qquad (2)$$

The actual disk stresses versus disk thickness for a typical small-diameter

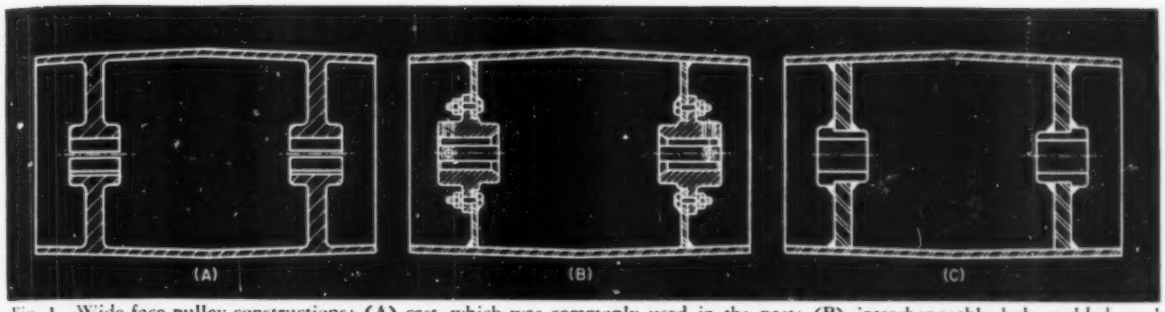


Fig. 1—Wide face pulley constructions: (A) cast, which was commonly used in the past; (B) interchangeable hub, welded steel, which can be adapted to any required shaft size; and (C) welded with solid steel hub, for heavy duty applications.

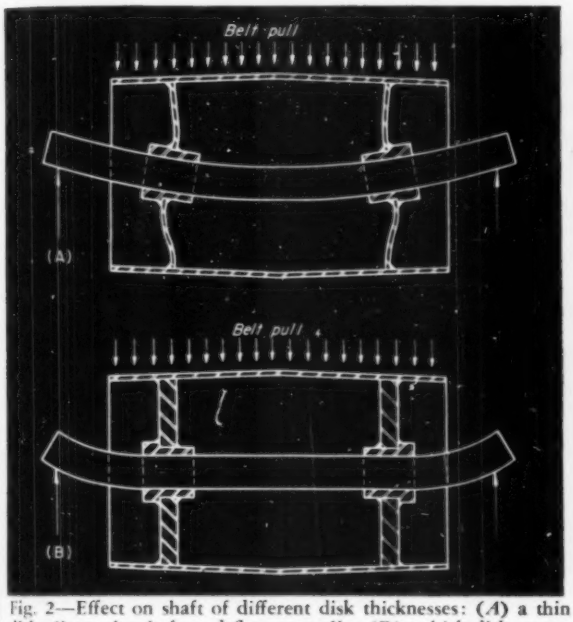


Fig. 2—Effect on shaft of different disk thicknesses: (*A*) a thin disk allows the shaft to deflect naturally; (*B*) a thick disk causes the shaft to deflect from the hub outward.

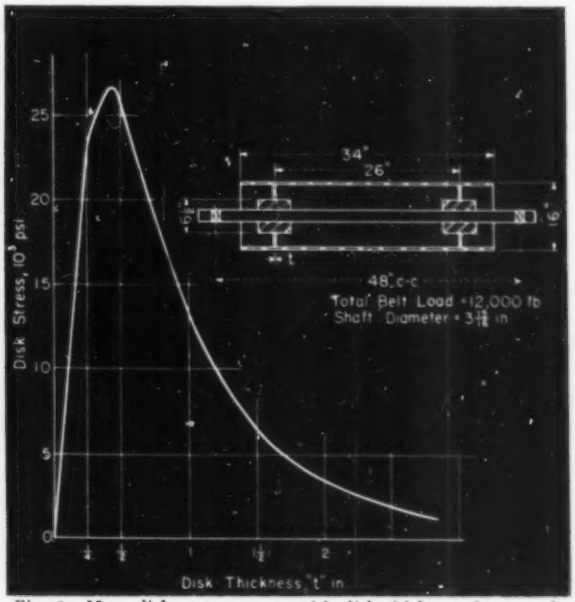


Fig. 3—How disk stresses vary with disk thickness for a typical small diameter wide face pulley. Note that the stresses increase, to a point, with increasing material thickness.

wide-face pulley is illustrated in Fig. 3. The stress increases almost in a straight line up to about $\frac{1}{4}$ in., and then increases at a lower rate until a stress of approximately 26,500 psi is reached at a thickness of about 0.4 inch. Beyond this point the stress falls off as the disk thickness is increased. Thus, for an allowable working fatigue stress there are two disk thicknesses that can be utilized: one a thin disk and the other a thick disk. Between these two limits a higher stress is inevitable. This represents one of the few cases in design where increasing a material thickness results in a higher, rather than in a lower stress in the part.

The disadvantage of employing thick disks to reduce stress is that the bending moment on the hub and therefore on the hub bolts is increased, and also that there is a tendency for a hub bore to wear and pound out, thus permitting the pulley to walk on the shaft. On small pulleys where the loads are light, rigid hubs are satisfactory, however.

To determine the effect of the location of the hubs or the end disks of a conveyor pulley, it is necessary to analyze Eq (1). If the two disks are close together, (X approaching zero), the angular deflection of the shaft at the point of hub location is small and therefore the disk stress is small; and if the disks are far apart (X approximating L), the shaft deflection is also small and thus the disk stress is small. Somewhere between these two extremities the shaft deflection for a given loading must reach a maximum. Differentiating this equation with respect to the X dimension reveals that this maximum occurs when X is equal to $\frac{1}{2}$ L. It so happens that in most pulley applications the logical location of the hubs, due to the length of bearings and frame clearances is close to this maximum point. When using Eq (1), it has been found that large diameter pulleys are seldom troublesome, but that small diameter pulleys, due to the increased stiffness of the disk, can very often be difficult to design. The reason for this is that in order to keep the disk stress below some practical values, say 10,000 psi, the disk thickness becomes impractically thin and, to carry the pulley load, it is necessary to revert to the thick-disk design. The swing from one design to the other is dependent mainly upon the magnitude of the total belt pull, P, and the center-to-center distance between bearings, L, as well as the center-to-center distance, X, between disks. A small shaft size resulting in a small moment of inertia also tends to raise the stress and sometimes it is necessary to specify a large shaft size to obtain a practical pulley design.

For standardizing a line of stock pulleys, it is necessary to supply appli-

cation engineers with some means of determining what shaft sizes must be used for various applications. It has been found practical to simplify Eq (1) by assuming that the pulleys always will be of the thin disk type and that the restraining effect of the disk on the shaft will be small. This results in safe shaft selections and greatly simplifies the formula that the prospective designer must employ.

The moment of inertia of the minimum shaft size can be calculated from the equation:

$$I_s = \frac{(P)\,(L - X)\,(X)}{K} \qquad (3)$$

in which K is the product of the allowable angular deflection of the hub, ϕ_s, the modulus of elasticity, E, and the constant, 8. The allowable angular deflection ϕ_s is given by the expression:

$$\phi_s = \frac{\delta D}{2\alpha \beta E t} \qquad (4)$$

Values of K can be readily determined for all sizes of pulleys. It is important to make certain that the hub bolts are strong enough to withstand the maximum bending moment that will be imposed on them, and that the direct radial stress in the disk in combination with the bending stresses do not exceed the fatigue limit of the materials.

Tests indicate that bolted hubs do not behave exactly as the Eq (1) would predict, but rather are somewhat more flexible and for a given angular deflection yield a somewhat lower stress. To correct for this error, a slightly higher disk stress is allowed and the bolt circle diameter is used as the hub diameter, d.

Also, Eq (2) does not take into account the stress in the shaft and it is therefore necessary to check shaft stress in all applications to make certain that the combined stresses in the shaft do not exceed a safe value according to the ASME code. Heavily loaded medium and narrow face pulleys are most apt to require attention because of shaft stress, whereas wide face pulleys are more apt to require larger shafts due to deflection.

Rim Thickness

Another element of the pulley that must be given a great deal of attention is the rim. Naturally it is necessary to employ thicker rims for heavy-duty pulleys than for light-duty pulleys, and it is natural to reason that there must be some relationship between the required thickness for a given belt pull, diameter of the pulley, and face width. These relationships are difficult to pin down, and it has only been as a result of a series of tests and experimental stress determinations that anything like a rational method of determining required rim thicknesses has been obtained.

For an infinitely long cylinder supported at either end, the stress is given by the formula:

$$S_t = \frac{0.157 \times T R}{t^2} \qquad (5)$$

where T is the belt tension on the tight side in lb/in. of face width; R is the pulley radius, in.; t is the rim thickness, in.; and S_t is the tensile stress, psi.

Thus the stress increases directly with an increase in belt pull and pulley diameter and decreases with an increase in rim thickness. Experimental tests on actual pulleys using a special strain gage, however, have not verified this equation, and it has been found that the rim deflection decreases with increase in pulley diameter for a given pulley width and rim thickness, the reason being that conveyor pulleys are actually not long cylinders but are short cylinders. It is possible to correlate the relationship between the spacing between disks and the diameter as these dimensions affect the stress in the rim for a given belt pull and rim thickness. The result of this work has provided an empirical method of computing rim thicknesses to meet the conditions of belt pull: by computing the rim thickness theoretically, using a practical allowable stress and then adding an allowance for rim wear. Of course, this allowance is not necessary when pulleys are lagged, but is decidedly necessary when pulleys are not lagged.

Small-diameter, wide-face pulleys with a large width to diameter ratio actually approach the conditions of an infinitely long cylinder. In such cases it is helpful to use a center disk, but this is only of value if it is properly installed. The deflections that occur in a pulley rim are very small and localized, and unless the center disk actually is a perfect fit it is possible for the rim stresses to develop in spite of the introduction of a center disk. To illustrate the reason for this, Fig. 4 has been drawn to show the nature of deflection that occurs in a pulley during operation. There is practically no change in the diameter under load, and yet the rim actually distorts and can develop high stresses as a given pulley section rolls from a point of no-load into a point under the belt where it is heavily loaded. The stress occurs at the transition point between no-load and full-load on the pulley. Thus any one section of the rim is subjected to two full reversals every revolution. Obviously if the rim is welded, as most of them are, it is the welded joint that is subject to the most suspicion. It is necessary to insure full weld penetration so that the joint does not have an inherent notch effect, resulting in a stress-raiser capable of causing failure prematurely.

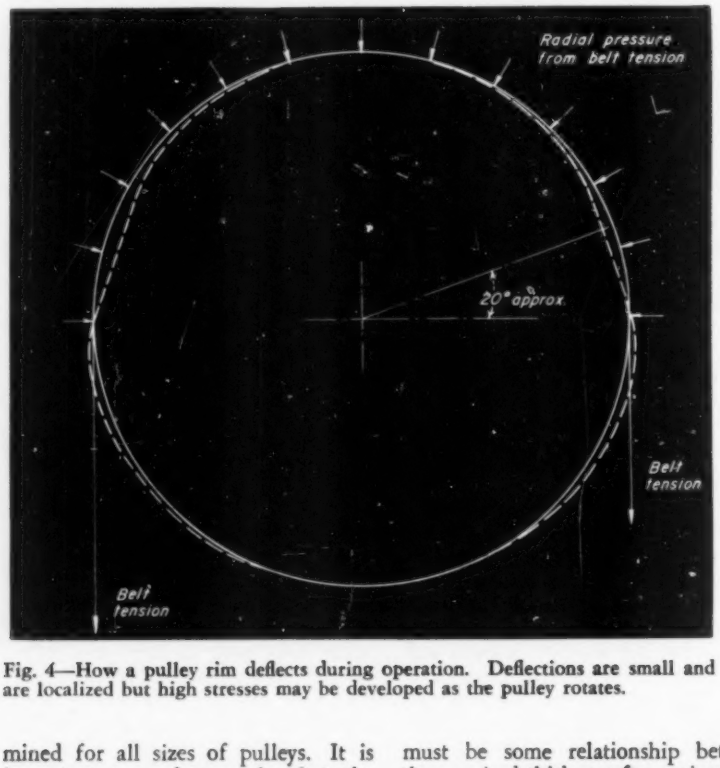
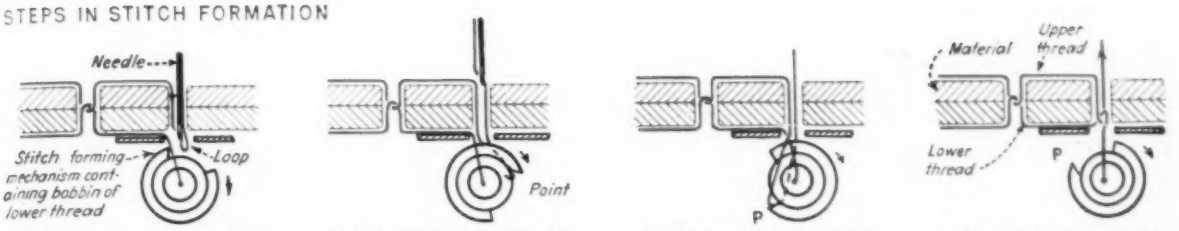


Fig. 4—How a pulley rim deflects during operation. Deflections are small and are localized but high stresses may be developed as the pulley rotates.

STEPS IN STITCH FORMATION



1 Needle pierces material and carries upper thread to the underside of material. When needle starts ascent, loop is formed in upper thread.

2 Point of stitch forming mechanism catches loop and carries it around. More thread is provided by the take-up mechanism for widening of loop.

3 Loop is carried around stitch forming mechanism and the lower thread, bobbin is inside the mechanism. The take-up mechanism starts retracting loop.

4 Stitch is finished when take-up has retracted all surplus thread and tightens the interlinked upper and lower thread in the middle of the material.

Fig. 1

Kinematics of Sewing Machines

Mechanical linkages for deriving periodic oscillating, reciprocating and rotary motions from a main shaft rotating at uniform angular speed. Basic mechanisms that are used in all home sewing machines are described and their characteristic kinematic advantages and disadvantages compared.

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SATISFACTORY PERFORMANCE of a sewing machine results from the properly timed operation of a number of component mechanisms. Each mechanism fulfills one special task. However different the various designs may be, the following component mechanisms are used on all models.

1. Stitch-forming mechanism.
2. Mechanism for imparting movement to the stitch forming mechanism.
3. Mechanism for deriving needle movement.
4. Upper thread take-up mechanism.
5. Tension
6. Presserfoot
7. Feed mechanism
8. Feed reverse and stitch-length-changing mechanism.

All home sewing machines use the lock stitch, Fig. 1, which is formed by an upper or needle thread and a lower or shuttle thread. These two threads are interlinked by passing the upper thread around the shuttle or bobbin containing the lower thread. The needle carries the upper thread to the underside of the material. The stitch-forming mechanism does the interlinking. The take-up mechanism in conjunction with the "tension" regulates the thread supply and tightens the stitch in the middle of the material. The feed mechanism moves the material by the length of one stitch after the stitch formation is finished. In all sewing machines, the number of stitches completed per minute is identical with the rpm of the main shaft, that is, one stitch is finished during one revolution.

STITCH FORMING MECHANISM. This mechanism interlinks the upper and lower thread, Fig. 1. The type of this mechanism differs according to the kind of interlinking member employed, which can be a shuttle or a hook; and according to the motion imparted to the interlinking member, which can be reciprocating, oscillating, or rotating.

A shuttle is not rigidly connected to the driving member. The shuttle is a separate piece held in place by ways or guides, and is driven by a shuttle driver merely by pushing. The shuttle contains the bobbin of lower thread and, as a rule, springs to provide the necessary tension for the lower thread. The shuttle is pointed at one end to catch the loop of the needle-thread.

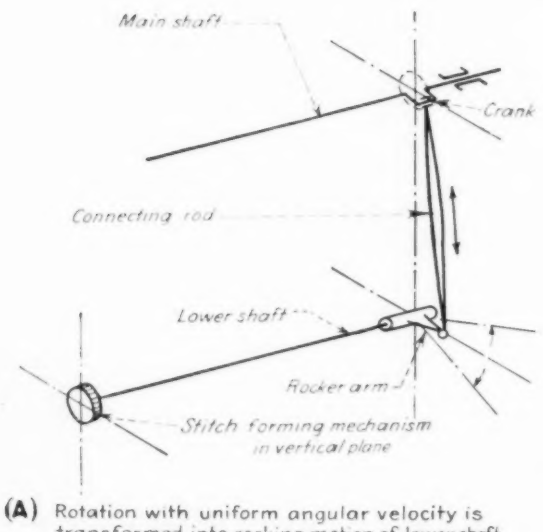
A hook is always in rigid connection with the driving shaft and, as a rule, has guides for holding the stationary bobbin of lower thread. A protrusion on the hook, called the point, catches the loop of thread and carries it around the stationary bobbin case.

In a reciprocating motion, the stitch-forming mechanism goes forward and backward in a straight line or in a sector of a circle. This motion is no longer employed on machines of modern design.

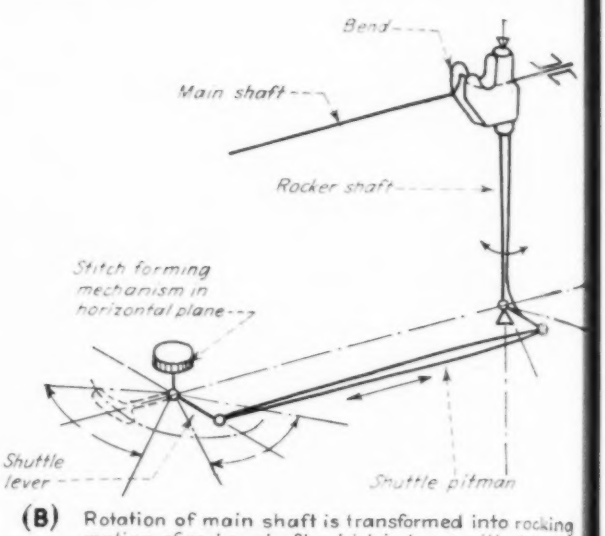
In an oscillating motion, the shuttle or hook swings around its axis forward and backward, usually with approximately 180 deg. angular displacement.

In a rotating motion, the shuttle or hook rotates around its axis with either a uniform or a variable angular velocity.

Reliable performance of the stitch-forming mechanism is essential for satisfactory operation of the machine. Whatever its principal design may be, the stitch forming mechanism must undergo a thorough test under all conditions of actual use, and all deficiencies must be corrected by empirical ad-

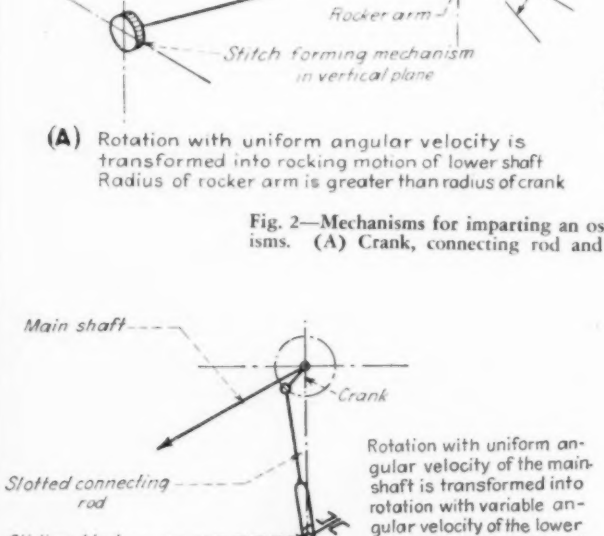


(A) Rotation with uniform angular velocity is transformed into rocking motion of lower shaft Radius of rocker arm is greater than radius of crank

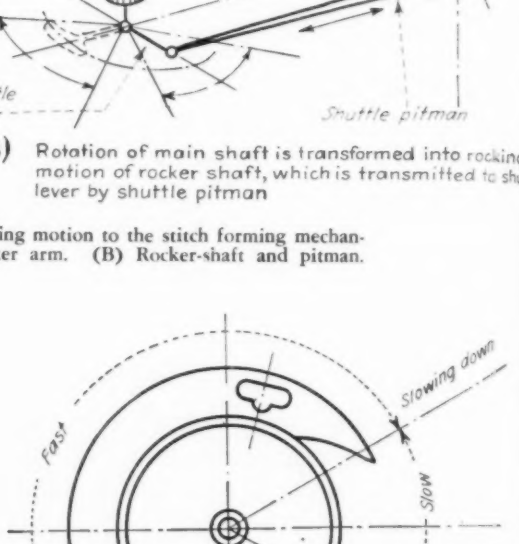


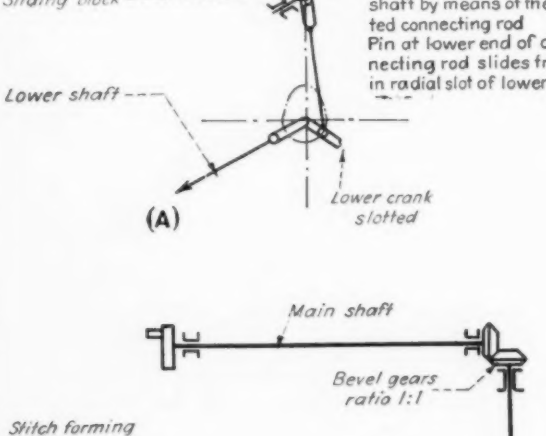
(B) Rotation of main shaft is transformed into rocking motion of rocker shaft, which is transmitted to shuttle lever by shuttle pitman

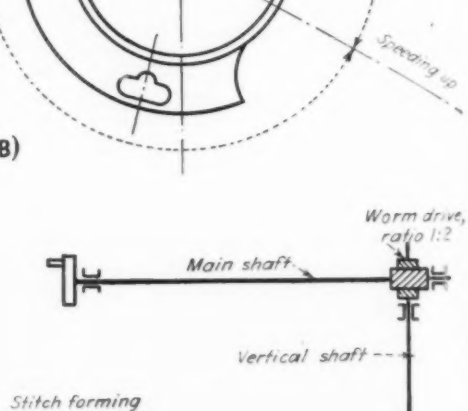
Fig. 2—Mechanisms for imparting an oscillating motion to the stitch forming mechanisms. (A) Crank, connecting rod and rocker arm. (B) Rocker-shaft and pitman.



Rotation with uniform angular velocity of the main shaft is transformed into rotation with variable angular velocity of the lower shaft by means of the slotted connecting rod
Pin at lower end of connecting rod slides freely in radial slot of lower crank







(D) Main shaft and lower shaft rotate at uniform angular velocity. Speed of stitch forming mechanism is twice that of main shaft.

Fig. 3—Mechanisms for imparting a rotary motion to stitch forming mechanism. (A) Crank and slotted connecting rod. (B) Variation in speed of shuttle or hook driven with slotted connecting rod. (C) Bevel gear transmission. (D) Worm drive transmission

justment before the mechanism is approved for manufacture. Once this status has been reached, all other mechanisms are built around the stitch forming mechanism.

In Figs. 2 and 3 are shown simplified diagrams of five principal types of mechanisms for imparting motion to the stitch forming mechanisms. The single feature they all have in common is that their motion is derived from the main shaft, which rotates at a uniform angular velocity.

The oscillating type with lower shaft, Fig. 2(A), employs a crank and rocker mechanism. The crank is fixed to the main shaft; the rocker arm to the lower shaft. A connecting rod, housed in the vertical portion of the machine connects the crank and rocker arm. Since the radius of the crank is smaller than the radius of the rock arm, the rotation of the main shaft is transformed into a rocking motion of the lower shaft.

The oscillating type with pitman, Fig. 2(B) employs a rocker-shaft as the transmitting member. A bend in the main shaft oscillates the rocker shaft about its vertical axis, and an arm at the lower end integral with the rocker shaft transmits the rocking motion by means of the shuttle pitman to the shuttle lever.

In Fig. 3(A) is shown a rotary type machine with a cyclic varying angular velocity of the lower shaft. The transmission for this drive is a combination of a "crank and slotted connecting rod" and a "pin and slotted crank." While the crank-pin of the main shaft crank describes a circle with a uniform angular velocity, the slotted connecting rod swings about the sliding block. Thus, the pin at the lower end of the connecting rod describes an egg-shaped curve with a variable velocity. This pin slides in the slot of the lower crank and drives it and the lower shaft connected with it with a variable angular velocity.

In Fig. 3(B) is shown how the point of a shuttle or hook, driven with the slotted connecting rod, varies in speed during one revolution. During 70 to 90 deg of angular displacement, it slows down to 0.28 of the main shaft speed, increases its speed, and then travels with a nearly constant stepped up angular speed 1.4 greater than that of the main shaft.

In Fig. 3(C) is shown a transmission of the rotary type with bevel gears, where the lower shaft rotates at a uniform angular velocity and is driven by a vertical shaft and sets of bevel gears.

The rotary type, shown in Fig. 3(D) is essentially the same as that in (C) except that a worm drive is used instead of bevel gears. In both types, the rotating hook stitch-forming mechanism rotates at double the revolutions per minute of the main shaft, making two revolutions for every stitch completed. During the first revolution the point of the hook catches the loop of the thread and carries it around the bobbin case. The take-up mechanism finishes the stitch and regulates the thread supply during the second revolution with the hook running idle. The angular speed of the rotating hook is uniform and the acceleration-deceleration cycles of the lower shaft are avoided.

Comparison of Oscillating and Rotating Machines

Oscillating and rotating types machines have been in use many years, and both have been developed to a high degree of perfection. However, it is obvious that under otherwise equal conditions (grade of material used, precision in manufacturing, duration and speed of operation, care in maintenance) a rotating type of machine will show less wear than an oscillating system.

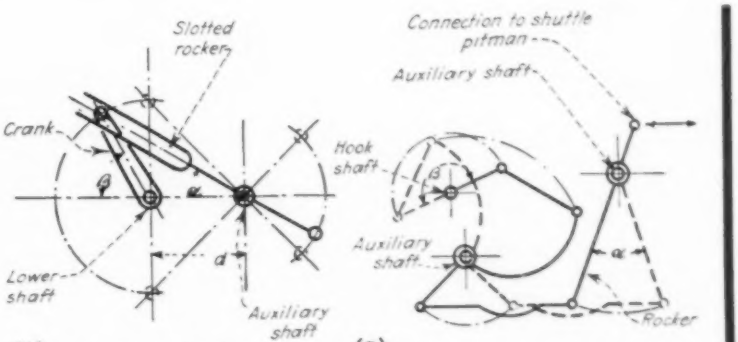
In the oscillating system, wear results from the inertia forces caused by the cyclic acceleration and deceleration of the oscillating parts, which must come to a complete stop twice during each cycle of stitch formation. The greater the mass of the oscillating parts, the greater, roughly speaking, are the inertia forces and bearing reactions responsible for wear, noise, and vibration. Designers of oscillating machines, of course, have recognized this fact and employed, as far as possible, mechanisms of smaller geometrical dimensions to decrease the weight of oscillating parts.

One low inertia design is shown in Fig. 4(A). This design resulted from an effort to reduce the inertia forces inherent in the mechanism shown in Fig. 2(A). The mass of the connecting rod between the crank of the main shaft and the rocker arm could not be reduced beyond a certain limit for reasons of strength. To obtain lower inertia forces, the solution was to reduce the angular displacement of the connecting rod. To maintain the required angular displacement of the lower shaft, an auxiliary slotted rocker and crank mechanism was used instead of the rocker arm.

Since the dimensions of the parts, Fig. 4(A), are small and their mass distribution close to their respective centers of rotation, in comparison with those of Fig. 2(A), the reduction in inertia forces was considerable.

In Fig. 4(B) is shown the solution of the similar problem of reducing the inertia forces present in the mechanism shown in Fig. 2(B). The angular displacement of the rocker shaft, was reduced and thus the displacement of the shuttle pitman. To maintain the required 180 deg displacement for oscillation of the hook, the shuttle lever was replaced by the double four link mechanism. This arrangement necessitated the addition of two auxiliary shafts.

The use of the mechanisms shown in Fig. 4(A) and (B) requires seven axles in the mechanism for imparting movement for stitch forming. For satisfactory operation, these axles must be parallel to each other and perpen-



(A) d = radius of crank

Angular displacement α of slotted rocker is transformed into an angular displacement β of the crank and lower shaft. β is twice α.

(B) Reciprocating motion of the shuttle pitman causes a rocking motion of rocker. Angular displacement α of rocker is transposed into an angular displacement β of the hook shaft by means of the double four link mechanism. β five times as large as α

Fig. 4—Low inertia designs. (A) Slotted rocker and crank. (B) Double four link mechanism.

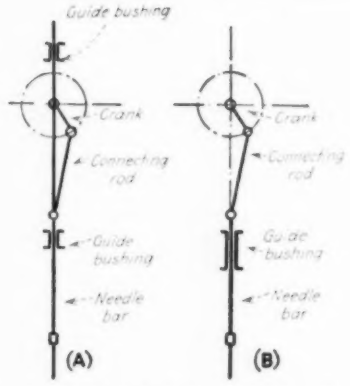
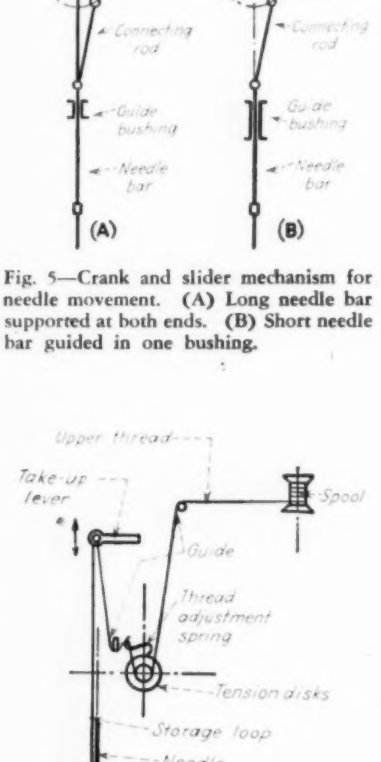


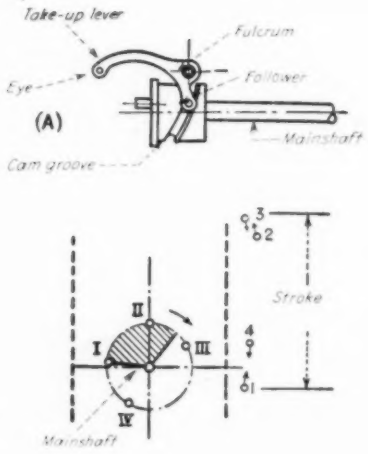
Fig. 5—Crank and slider mechanism for needle movement. (A) Long needle bar supported at both ends. (B) Short needle bar guided in one bushing.



(a) Shuttle point catches loop of upper thread. Take-up lever moves downward to supply thread for widening of loop while carried around shuttle

(b) After loop has been carried around shuttle, take-up lever has to retract surplus of thread in a short time (30 percent of cycle) to finish interlocking of stitch inside the material and to prevent shuttle point catching same loop a second time

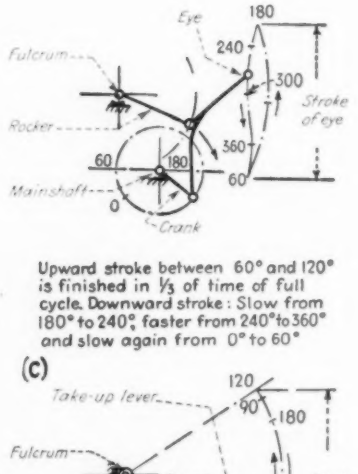
Fig. 6—(A) Threading of upper thread. Disks provide most of the thread tension. (B) Regulation of thread supply by take-up during one cycle of stitch formation.



I, II, III, IV Positions of crank
1, 2, 3, 4 Corresponding positions of eye of take-up lever
S = Stroke of take-up lever

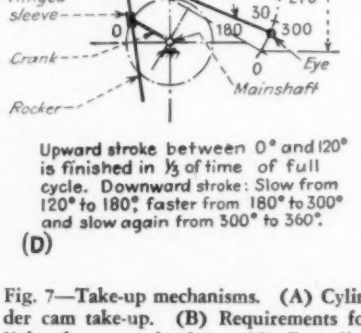
Shaded portion indicates angular displacement of crank, during which upward stroke of take-up lever must be finished. As angular velocity of crank is constant, upward stroke is fast; downward stroke slow, speeding up slightly between III and IV or 3 and 4 respectively.

(B)



Upward stroke between 60° and 120° is finished in ⅓ of time of full cycle. Downward stroke: Slow from 180° to 240°, faster from 240° to 360° and slow again from 0° to 60°

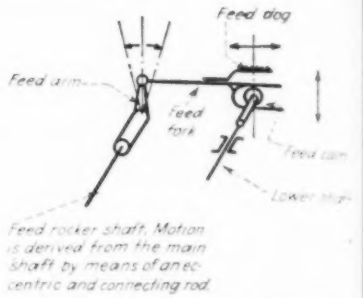
(C)



Upward stroke between 0° and 120° is finished in ⅓ of time of full cycle. Downward stroke: Slow from 120° to 180°, faster from 180° to 300° and slow again from 300° to 360°.

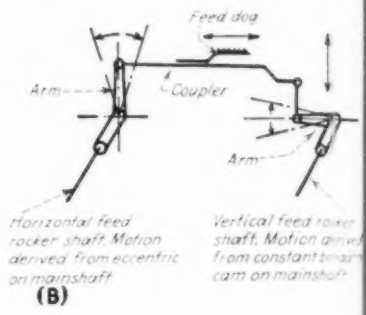
(D)

Fig. 7—Take-up mechanisms. (A) Cylinder cam take-up. (B) Requirements for link take-up mechanism. (C) Four link take-up. (D) Crank and rocker take-up.


Feed dog
Feed arm
Feed fork
Feed cam
Lower shaft
Feed rocker shaft. Motion is derived from the main shaft by means of an eccentric and connecting rod

Feed rocker shaft performs a rocking motion, from which the horizontal feed motion is derived.
Feed cam revolves at the variable angular velocity of lower shaft and causes the vertical movement (up and down) of feed dog.

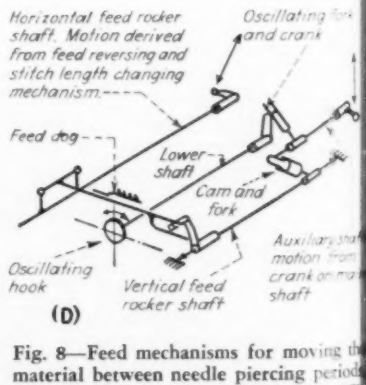
(A)


Feed dog
Arm
Coupler
Arm
Horizontal feed rocker shaft. Motion derived from eccentric on mainshaft
Vertical feed rocker shaft. Motion derived from constant breadth cam on mainshaft

(B)

Feed dog
Arm
Forked lever
Horizontal feed rocker shaft. Motion derived from constant breadth cam on mainshaft
Vertical feed rocker shaft. Motion derived from eccentric on main shaft

(C)


Horizontal feed rocker shaft. Motion derived from feed reversing and stitch length changing mechanism
Oscillating fork and crank
Feed dog
Lower shaft
Cam and fork
Oscillating hook
Vertical feed rocker shaft
Auxiliary shaft motion from crank on main shaft

(D)

Fig. 8—Feed mechanisms for moving the material between needle piercing periods. (A) For differential machines. (B), (C), (D), For oscillating machines.

dicular to the main shaft. Hence, manufacturing problems were increased by this arrangement. Decreasing the tolerance limits increase the cost of manufacture, while liberal clearance allowances result in noisy operation after a short period of service.

It was an improvement, therefore, when the rotary type machine Fig. 3(A) was developed. Here the shuttle or hook rotates, though its speed varies, without stopping during the cycle of stitch formation. The velocity of the point of the hook is smallest when the needle reaches the lowest point, that is, shortly before the point of the hook and needle meet. The hook speeds up and travels at a high speed while the loop of thread passes around the bobbin case, and then slows down to give the take-up mechanism sufficient time to retract the surplus thread, tighten the stitch, and provide enough thread for the formation of the loop for the next stitch.

This differential drive was succesful in solving the difficulties associated with a rotary shuttle or hook, and is relatively free of vibrations when operated at speeds up to 600 rpm. For sewing machines driven by foot power a speed of 600 rpm is satisfactory. With the introduction of power driven sewing machines, speeds ranging from 900 to 1300 rpm have come into use. Here the inertia forces, caused by the heavy connecting rod swinging up and down and rocking around the stationary fulcrum, and by the accelerations and decelerations of the rotating member, tend to produce strong mechanical vibrations.

For the modern power driven home sewing machine operating at speeds as great as 1300 rpm, the gear driven rotary hook designs, Figs. 3(C) and (D), present an adequate solution of the problem. In these designs, oscillating parts are avoided as much as possible; consequently vibration and noise in operation are reduced.

Needle Movement and Thread Supply

Though all home sewing machines use the crank and slider mechanism for deriving the reciprocating movement of needle bar from the rotation of the main shaft, there are two different types as shown in Figs. 5(A) and (B). In (A), a long needle bar is supported on both ends of the casting; while in (B), the needle bar is short and is supported only on one side in a hardened steel bushing.

Analytical considerations show that the reaction to the inertia force of the moving needle bar tends to excite a forced vibration of the head of the machine.

Without reducing the speed, the reaction could be reduced by decreasing the crank radius r, but the total stroke (equals $2r$) of the needle and its velocity determines the length of radius r. For a required stroke, therefore, the reaction can be reduced within certain limits only by decreasing the mass of the needle bar.

It is evident that supporting the needle bar only in a short bushing presents a disadvantage. The driving connecting rod causes a couple acting on the needle bar, varying with the load (friction in the bushing, friction between needle and thread and resistance the needle finds when piercing the material) and the angle. This couple must be counteracted by another one, which produces corresponding reactions in the sleeves. These reactions are responsible for wear in the bushing, and they increase as the length of the bushing becomes shorter. In manufacture the alignment of this bushing presents other difficulties, but modern production methods and the use of hardened steel bushings have overcome these disadvantages.

MECHANISMS REGULATING THE THREAD SUPPLY; TAKE-UP AND TENSION. For each stitch, the thread supply must yield a certain length of upper and lower thread, approximately equal to the length of the stitch and keep the threads under such a tension that the interlinking of the two threads is firm and tight. It is relatively simple to obtain tension for the lower thread by springs on the shuttle or bobbin case, adjustable by means of a screw driver. For the upper thread an additional device must be provided to furnish sufficient thread for widening the loop of thread when carried around the bobbin, and to retract the surplus when the stitch is tightened.

The threading of the upper thread is shown in Fig. 6(A). The tension disks provide the tension; the thread adjustment spring supplies and additional tension to compensate for slight inaccuracies in adjustment. After passing the eyelet of the take-up lever, the thread forms a storage loop, the length of which varies with the position of the up-and-down moving eyelet of the lever.

In Fig. 6(B) is shown how the thread supply is regulated. The amount of thread to be furnished for the widening of the loop that passes around the shuttle, as well as the timing of supply and take-up is entirely dependent on the shape and kind of movement of the stitch-forming mechanism.

There are two principal kinds of take-up mechanism: (1) the cylinder cam take-up, and (2) the four-link take-up. Fig. 7(A) shows the cylinder cam take-up mechanism. A first class lever with its fulcrum in the casting of the head of the machine, has a follower pin fastened to one end. This pin slides in the groove of the cylinder cam that is fastened to the main shaft. By correct choice of cam groove, movement of the eye of the take-up lever can be made to suit requirements.

The requirements of a four-link mechanism that must be fulfilled are shown in Fig. 7(B). One solution of the problem is shown in Fig. 7(C). In Fig. 7(D) is shown a solution for a crank and rocker type of take-up.

Among the take-up mechanisms, the cylinder cam take-up offers the designer the greatest freedom in adjusting the movement of the eye of the take-up lever to the requirements of the stitch-forming mechanism. The cam groove on the cylinder can be easily generated to suit the required motion. Incidentally, all the cam grooves were developed empirically over a period of years. In many instances, they were copied from older models without any critical analysis.

The disadvantage of the cam type is that it is either noise producing or power consuming. If groove and follower are machined to close tolerances, there is considerable loss of power because of friction; further, binding may occur when the relative positions of cam and lever are not kept within close tolerances. When the follower is loose in the groove or the groove is worn through use, a strong metallic noise of bothersome high frequencies is produced, especially at high speeds.

The main reason why link take-ups are rarely used is the difficulty in obtaining displacement curves identical to those generated by a cam. In many designs, however, curves obtained by four-link mechanisms can be used. The advantages of a four-link take-up are: Low inertia forces, low braking effect on the machine, wears well, and comparatively is less noisy.

The type shown in Fig. 7(C) causes some difficulties in production, as again a number of axles have to be kept parallel to each other and the clearance in the links should not exceed a certain amount. The type shown in Fig. 7(D), is claimed to be easier to produce and therefore cheaper; it has the kinematic advantage that the eye describes an arc of a circle, without bending it on the upward stroke.

Feed Mechanisms

After the needle has pierced the material and the interlinking of the upper and lower thread has been performed, it is necessary to move the material a certain distance for the next piercing

of the material with the subsequent thread interlocking.

This movement of the material is done by the feed dog in conjunction with the presserfoot. While holding the material down on the needle plate with a yielding spring tension, the toothed presserfoot rises several thousandths of an inch above the needle plate to enable the teeth of the feed dog to grip the material. Then the feed dog moves a certain distance in the direction of the feed, taking the material along, and then drops below the needle plate to move back to the starting position.

This kind of feed is called a four motion feed, according to its more or less distinct up and down forward and backward movements.

On machines with differential movement of the lower shaft, Fig. 3(A) the up-and-down movement of the feed dog is derived from a cam on the lower shaft, Fig. 8(A). On machines with constant speed of the lower shaft, Figs. 3(C) and (D); or oscillating machines, Fig. 2(A); the up-and-down movement is derived from the upper shaft by means of a crank or cam in conjunction with a rocker, Figs. 8(B), (C), and (D), below the bed of the machine. In Fig. 8(D), the oscillating fork and crank doubles the angular displacement of the lower shaft with respect to the auxiliary shaft.

The mechanisms shown in Figs. 8(B) and (C) differ inasmuch as the interconnection of the up-and-down and back-and-forward motion of the feed dog is concerned. In Fig. 8(B) a short link is used: in Fig. 8(C) a fork and pin connection is employed. Since the fit between an axle and an eye of the lever can be worked to close tolerances, the short link type promises to operate with less noise than the fork and pin connection.

FEED REVERSE MECHANISM AND CHANGING OF LENGTH OF STITCH. To reverse the direction of sewing, the feed dog has to transport the material in the opposite direction, while the up-and-down movement of the feed dog remains unchanged.

There are two ways of accomplishing this movement. The feed motion shown in Fig. 9(A) is derived from an eccentric in the main shaft. Below the bed of the machine is a circular disk with a diagonal slot that can be turned through 90 deg, so the slot can be brought into all intermediate positions. The connecting rod has at its lower end a pin that fits into the slot of the disk. As the main shaft turns one revolution, this pin slides in the diagonal slot down and up. A short lever transmits this motion of the pin to the feed rocking lever and the feed rocker shaft, causing the rocking motion for the feed movements. By turning the slot into a symmetrical position, the direction of the feed will be reversed and maintain the same length of stitch. Intermediate symmetrical positions between the two extremes will permit any stitch length down to zero in opposite directions.

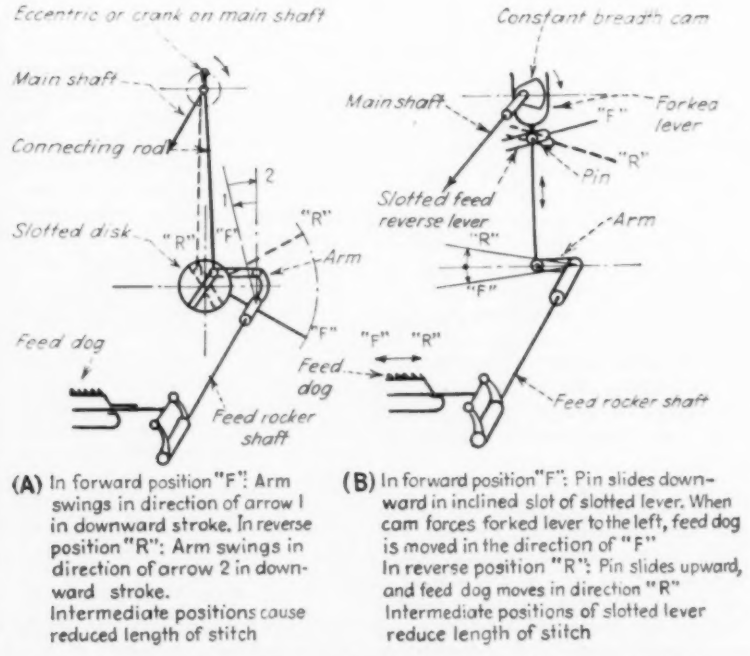
The feed reversing and stitch length changing mechanism shown in Fig. 9(B) employs a constant breadth cam in conjunction with a forked connecting rod, linked at the lower end to the feed rocking lever. A pin in this connecting rod, below the fork, slides in a slot that can be adjusted in various positions.

The constant breadth cam is usually designed to give a harmonic lateral displacement from 0 to 90 and 180 to 270 deg. angular displacement of the main shaft, and no displacement from 90 to 180 and 270 to 360 deg. During the motion period the pin is forced up and down in the inclined slot, causing the rocking motion of the feed rocking lever. Turning the inclined slot into a symmetrical position reverses feed.

The constant breadth cam has the advantage of limiting the feed motion of the feed dog to only 25 percent of the time of one full revolution. By using eccentrics the forward and backward motions overlap with the up and down motions, causing more or less ellipitically shaped displacement time diagrams of the feed dog instead of the ideal rectangle-shaped displacement.

Comparing the mechanisms of Figs. 9(A) and (B) the constant breath cam in the main shaft offers the advantage of moving the feed dog in the feed direction and back in only two quarter turns of the main shaft, and also permits the dog to stay at rest during a quarter turn each time in between. The up-and-down motion is generally a harmonic motion, derived from an eccentric; the mechanism of Fig. 8(A) is an exception, since the variable speed of lower shaft distorts the otherwise harmonic motion.

The displacement diagram of the feed dog of an ideal feed mechanism is a rectangle; that is, the up-and-down and forward-and-backward motions of the feed dog do not overlap. While this ideal condition is not realized in any of the feed mechanisms, their operation is not affected too seriously. It is essential, however, that as long as the needle is below the needle plate that the material remain at rest. During this period the feed dog should stay at rest or stay below the needle plate, so that movements of the dog do not affect the material being sewed.



(A) In forward position "F": Arm swings in direction of arrow 1 in downward stroke. In reverse position "R": Arm swings in direction of arrow 2 in downward stroke. Intermediate positions cause reduced length of stitch

(B) In forward position "F": Pin slides downward in inclined slot of slotted lever. When cam forces forked lever to the left, feed dog is moved in the direction of "F" In reverse position "R": Pin slides upward, and feed dog moves in direction "R" Intermediate positions of slotted lever reduce length of stitch

Fig. 9—Feed reversing and stitch length changing mechanisms. (A) Eccentric in main shaft and slotted adjustment disk. (B) Constant breadth cam.

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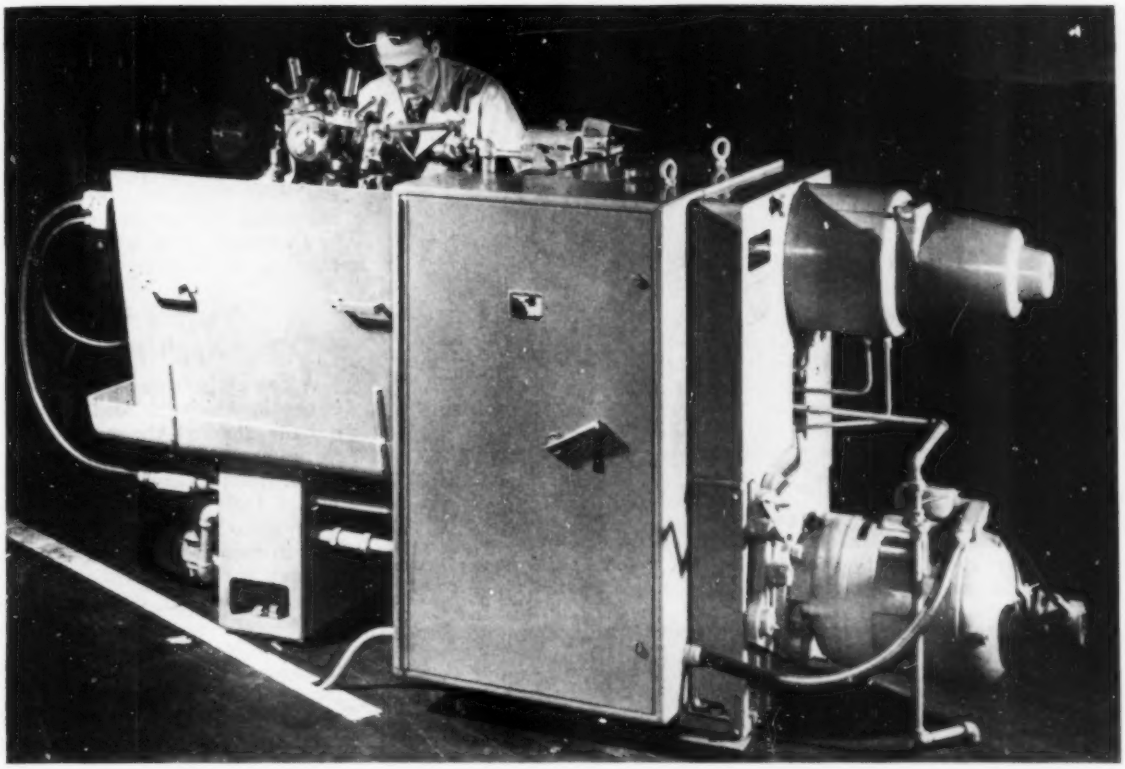


SEVERE DUTY CYCLES are frequently met in machine tools. Designers must consider the heating effect of this type of service before selecting a drive motor. Warner and Swasey No. 3 Electro-Cycle Turret Lathe styled by Henry Dreyfuss.

L. H. BERKLEY
Application Engineer
The Master Electric Company

Determination Of

Motor Horsepower From Duty Cycle

THE SELECTION OF THE PROPER motor rating for an electrically driven machine is comparatively simple when the machine is to operate continuously under a uniform load. It requires merely the determination of continuous running horsepower and the starting torque needed. However, many present-day automatic and semi-automatic machines, machine tools for example, involve complicated operations that impose non-uniform loads on the driving motor. Some motors are subject to varying loads while running continuously, while others must accelerate or plug-reverse frequently, perhaps under heavy inertia loads. Such duty cycles are classified as severe because of the heat load they place on the drive motor. When designing for severe duty cycles, the engineer must consider this additional factor of heating or face possible motor burnout.

The basic problem, then, is to select a motor capable of delivering the torque requirements of the load without excessive temperature rise. At the same time, the machine must not be over-motored, for this is accompanied by excessive first cost, and higher operating expense that is caused by lower efficiency and power factor. Many times the motor is selected on the basis of past experience, usually with a few safety factors, such as building it in an oversize frame or with a high-slip or lightweight rotor. These safety factors, which may or may not be necessary, invariably increase the cost of the motor.

Fortunately, there is a more accurate technique for selecting motors for

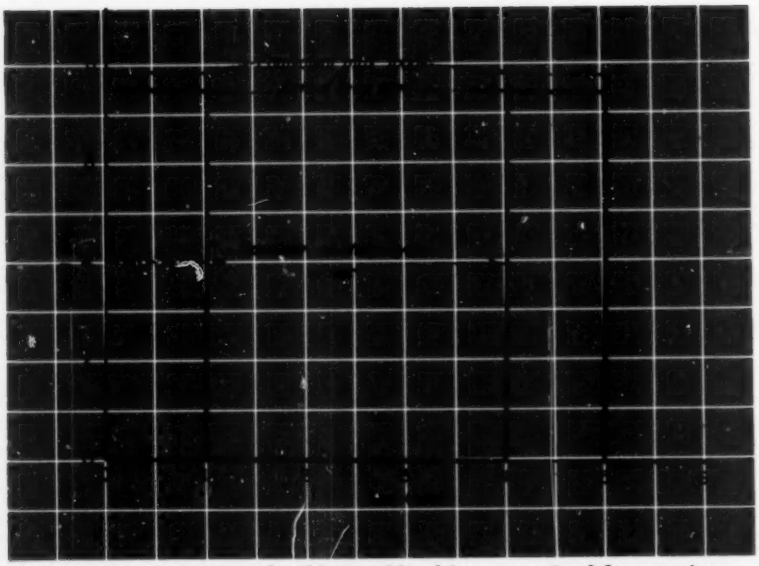


Fig. 1—Duty cycle for a woodworking machine drive motor. Load fluctuates between idle and 250 percent rated. Equivalent max load has been adjusted for speed drop.

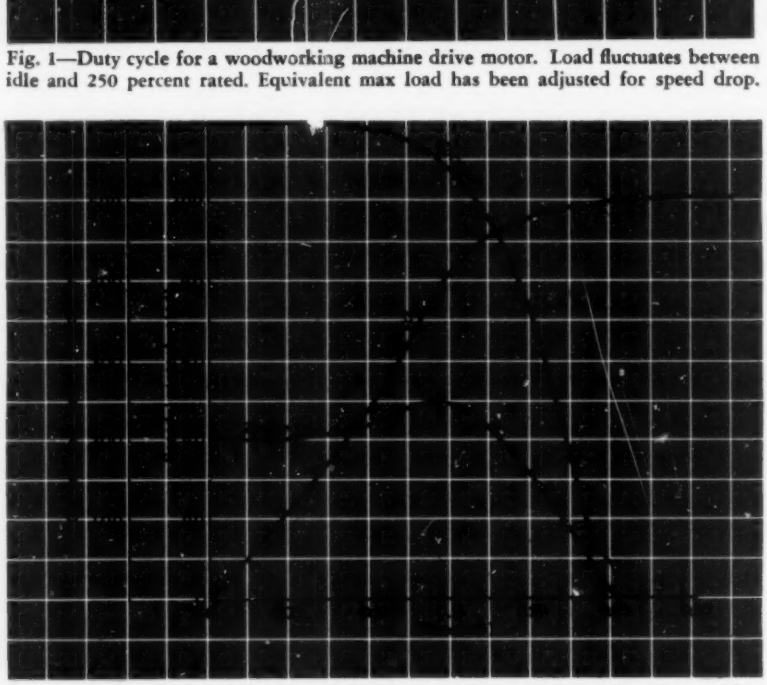


Fig. 2—Characteristic curves for a polyphase squirrel-cage motor. Frequent acceleration affects the load capacity of this motor as shown by current versus time curve.

severe duty cycles. This involves calculating the equivalent horsepower of the duty cycle and then choosing a motor of sufficient rating to deliver that power continuously without overheating. The equivalent value of a varying hp load is defined as that continuous value which has the same total heating effect as the varying load.

Equivalent horsepower can be calculated by several methods. One of the simplest is to determine the equivalent horsepower during each part of the duty cycle. Then, by taking the root-mean-square of these values, the designer arrives at equivalent horsepower for the entire cycle.

The equivalent horsepower method is applicable only to those motors whose speed regulation is fairly low. But since these include most single- and three-phase induction motors, and most shunt or compound d-c motors, the method can be used for probably 95 percent of all applications. The method gives only a fair degree of accuracy with enclosed motors and is not at all applicable to series motors.

Motor heating at any load is determined by in-phase current input. In motors of low speed regulation, this current is almost directly proportional to the developed torque. Hence, the square of the developed torque is a measure of heating (I^2R). Inasmuch as speed is fairly constant, the square of the horsepower developed at a particular instant is also a measure of heating. Therefore, the equivalent heating—or equivalent horsepower—during the entire duty cycle can then be determined from the root-mean-square of the equivalent horsepower of the individual periods, just as the heating caused by a varying current can be determined from its rms values.

Varying Load Application

A duty cycle frequently encountered is one in which the load varies periodically, although the motor continues to run at a fairly constant, near rated speed. Such a duty cycle is shown in Fig. 1 (solid lines), for the cutter-drive motors of a hand-hole cutter, used in manufacture of wooden boxes. As each board reaches the cutters, the load rises rapidly. Between each operation, the load is practically zero.

Normally the machine employs a three hp, 3,450 rpm motor. However, the wood in the vicinity in which this particular machine was located had a tough grain and was extremely difficult to cut. The speed of the motors decreased rapidly during the cutting operation and the motors were excessively hot. After failure of one of the motors, it was decided to use more powerful units. The question arose as to the required rating.

In checking the load during actual operation, it was found that actual load peak was approximately 7.5 hp or 250 percent rated. The actual duty cycle shown by the solid lines of Fig. 1 was then constructed and the equivalent rms horsepower calculated.

In calculating rms horsepower, one must be mindful that heating is determined by delivered torque or equivalent horsepower. Because the motor speed is somewhat lower at a load of 250 percent rated (7.5 hp) than at rated (three hp), the torque at 7.5 hp is more than 250 percent rated. Consequently, the heating at 250 percent rated load is more than 250 percent of the heating at rated load—and so is the equivalent horsepower. This is borne out by the fact that efficiency in a normal motor is lower at overload than at rated load and hence heating at overload is in excess of the simple percent of overload.

Because the ratio between heating at overload and heating at rated load is proportional to torque, it is inversely proportional to speed. Consequently:

$$\text{Equivalent Peak } HP = \text{Actual Peak } HP \frac{\text{Speed at Full Load}}{\text{Speed at Overload}} \quad (1)$$

This adjustment of equivalent horsepower for speed is not made for less than full load, where speed is above rated, even though efficiency is less than rated. In duty cycles such as this, where loads vary from no load to several times rated, the slight additional accuracy gained by this refinement does not generally warrant the additional calculations.

The speed of the motor at rated load is 3,485 rpm; the speed at 250 percent overload is 3,325 rpm. From Eq (1):

Equivalent Peak $HP = 7.5 \frac{3485}{3325}$ or 7.85 hp

The equivalent horsepower duty cycle is then constructed as shown in Fig. 1 by the dotted line. The rms horsepower that the motor must carry during the entire duty cycle can then be calculated.

Determination of rms horsepower is based on the definitions:

$$HP_{rms} = \sqrt{\frac{1}{t}\int_{0}^{t} (HP)^2\, dt} \qquad (2)$$

Where t = the duration of the duty cycle.
HP = the function of equivalent horsepower during the duty cycle.

For a motor running continuously at approximately constant speed, the formula can be expressed in more easily applied terms:

$$HP_{rms} = \sqrt{\frac{1}{t}\left[(HP_1)^2 t_1 + (HP_2)^2 t_2 + (HP_3)^2 t_3 \ldots . (HP_n)^2 t_n\right]} \qquad (3)$$

Where t = the duration of the entire duty cycle.
HP_1 = the equivalent horsepower during the first period of the duty cycle.
t_1 = the duration of the first period.
HP_2, HP_3, HP_n, t_2, t_3, and t_n represent the second, third and subsequent periods.

Applying this equation to the equivalent duty cycle of Fig. 1:

$$HP_{rms} = \sqrt{\frac{(7.85)^2\, 1 + (0)^2\, 3}{4}} = 3.92$$

This calculation indicates that the load imposed by duty cycle is equal to 3.92 hp continuously. In other words, the three hp motor was subjected to a continuous overload of over 30 percent. Also, and equally important, the peak torque required is close to the breakdown torque of the three hp motor.

The next step was to select a motor capable of performing the duty cycle. Because of interchangeability restrictions, the motor had to be built in the same frame diameter as the three hp motor. Were the next larger rating 4 or 5 hp, it could be safely recommended. However, the next larger rating available in this frame diameter is a 5 hp 60 min intermittent motor.

To determine whether or not a 5 hp, 60 min motor will be satisfactory, it must be noted that a 60 min motor will generally deliver from 70 to 80 percent of its rating continuously. Consequently, this motor has an equivalent continuous rating of approximately 0.75 x 5.0 or 3.75 horsepower. The duty cycle continuous load of 3.92 hp therefore represents an overload of only four percent and consequently the 5 hp, 60 min rating will be satisfactory. Also, since a 5 hp motor will run at a higher speed during peak load, the equivalent peak horsepower will be less than 7.85 hp, thereby reducing the rms horsepower.

This case is typical of those in which intermittent-duty motors are applied. The intermittent motor is often used where the duty cycle imposes a high ratio of maximum torque to average torque. Consequently, to build a motor in the required frame size and capable of delivering the required torque, it is often necessary to sacrifice heating capacity by using an intermittent rather than a continuous rating. But torque or horsepower capacity is gained in the exchange. This insures that the intermittent motor has sufficient maximum torque to supply

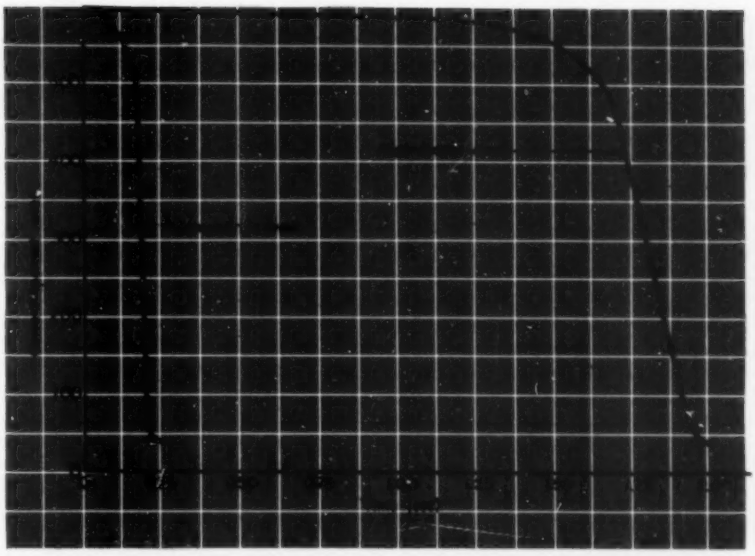


Fig. 3—Time required for acceleration is directly related to inertia of load. Thus, inertia has an effect on duty cycle because of higher input currents during acceleration.

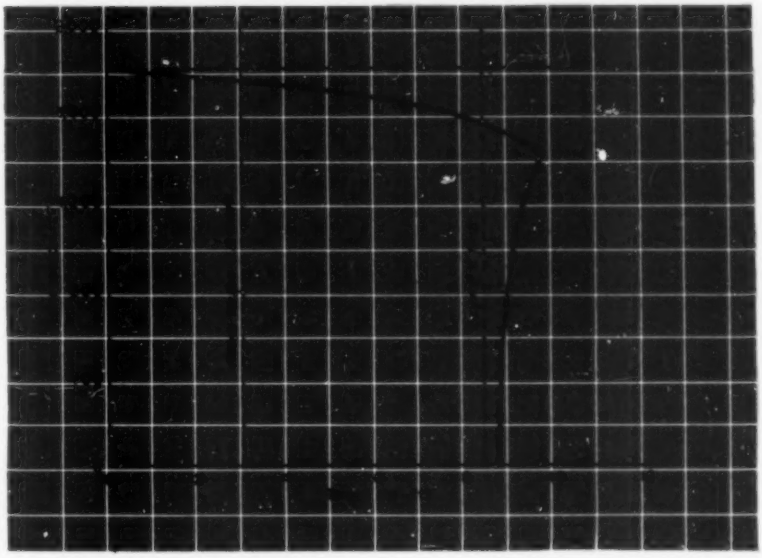


Fig. 4—Speed-torque curve for a one hp, 1,725 rpm polyphase motor. Heating effect during acceleration depends on the average torque of the motor during that period.

the peak loads without pulling out. An intermittent motor will often perform more satisfactorily than will a continuous motor of lower rating in the same frame. The price of the intermittent motor in ratings larger than about 20 hp at 1,750 rpm will actually be lower than the next lower continuous rating.

In applying intermittent motors to such applications, it must be remembered that most 30 min motors, because of high magnetization of the iron and consequent high no-load losses, cannot run continuously, even at no load, without overheating.

Another possibility to be considered in applying motors for continuous-running varying-load duty cycles is the Design-D, high-torque, high-slip motor. Such motors are particularly advantageous where flywheels are used to reduce peak load and light flicker. Design-D motors have a flatter line current vs torque curve than Design-B motors.

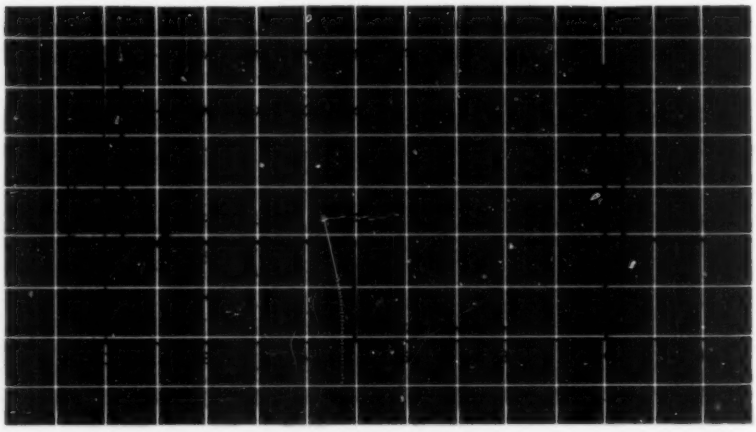


Fig. 5—Equivalent horsepower curve for a conveyor that is subject to frequent starting and stopping. Heating effect on drive motor during each start is considerable.

Frequent-Starting Application

Before analyzing a frequent-starting application, consider first what happens to a standard NEMA Design-B polyphase squirrel-cage motor when it accelerates without load. From Fig. 2 it is seen that the induction motor alone comes to rated speed in about ¼ second. With locked-rotor, current is about 6 times rated and decreases as the motor accelerates. During this starting period, the heat generated within the motor is many times that generated during normal operation at full load. Thus, the load-carrying capacity of a motor is limited by the frequency of accelerations.

Another factor to be considered is inertia. As indicated by Fig. 3, the larger the inertia, the longer is the time required for acceleration. Consequently, the total I^2Rt losses and heating are proportionately greater for high inertia loads. It is apparent from the above that heating is dependent on the frequency of accelerations or plug-reversals and on inertia. The problem is to calculate the relative effect of these on motor rating.

The torque delivered by a motor during acceleration depends on its speed-torque curve, an example of which is Fig. 4 for a NEMA Design-B, one hp, 1,725 rpm polyphase motor. The average torque during acceleration is approximately 8.4 lb-feet. Since heating is dependent on delivered torque, the heating during acceleration is approximately identical to that if the motor were delivering 8.4 lb-ft continuously at rated speed. This analysis neglects, for the moment, the consideration that speed during acceleration differs from rated speed and is not constant. The adjustment for speed is discussed later. The equivalent horsepower during the accelerating period is then:

$$HP_e = \frac{T_{avg} \times RPM}{5{,}252} \qquad (4)$$

Where T_{avg} = average torque in lb-ft delivered by motor during the accelerating period
RPM = rated motor speed, rpm

In the case of the one hp, 1,725 rpm Design-B motor:

$$HP_e = \frac{8.4 \times 1{,}725}{5{,}252} = 2.76$$

Heating during the accelerating period is the same as if the motor were delivering 2.76 hp for the duration of the accelerating period. It should be noted that:

$$HP_e = \frac{T_{avg}}{T_{rated}} \; HP\ rated \qquad (5)$$

A typical problem involves a conveyor that must start every 10 sec (6 times per min), run 5 sec, then stop. It is idle during the remaining 5 seconds. The problem is to find out whether or not a standard NEMA Design-B motor will do the job without overheating.

For steady running service on other conveyors of the same type, the manufacturer employs a polyphase, right angle, worm gearmotor rated one hp, 1,725 rpm motor speed, 58 rpm countershaft speed (see Fig. 4, torque-speed curve). Consequently, it can be assumed that after the motor reaches full speed, the load during the remainder of the 5 sec running period will not exceed one horsepower. Knowing the load on the conveyor and its linear speed, its effective inertia when referred to the motor shaft can be calculated by the formula:

$$\text{Effective } WR^2 = W\left(\frac{V}{2\pi S}\right)^2 \text{ lb-ft}^2 \qquad (6)$$

Where W = Weight of moving load, including the conveyor, lb
V = Velocity of conveyor, feet per minute
S = RPM of motor shaft

If the load is in rotary instead of linear motion, the effective WR^2 referred to the motor shaft is calculated by:

$$\text{Effective } WR^2 = \text{Actual } WR^2 \text{ of Load}\left(\frac{\text{Load } RPM}{\text{Motor } RPM}\right)^2 \qquad (7)$$

However, in this case, the inertia of the load is negligible when referred and compared to the rotor inertia because of the slow conveyor speed. Accelerating time is calculated as follows:

$$T_{acc} = \frac{WR^2 S}{308\, t_{acc}} \qquad (8)$$

Where t_{acc} = time in sec
WR^2 = total WR^2 of the load when referred to the motor shaft, lb-ft²
S = rated speed of the motor, rpm
T_{acc} = average torque *available for acceleration*, lb-ft

The average torque available for acceleration is not equal to the average delivered by the motor. During acceleration, the motor must overcome running friction of the bearings and gears, i.e., the friction that imposes the one hp load during steady running conditions. Since the steady running load is one hp, the equivalent friction torque at the motor shaft is equal to rated torque or 3.0 lb-feet. Consequently, the average torque available for acceleration (T_{acc}) is approximately equal to the average torque delivered by the

motor (T_{avg}) minus the friction torque. Therefore:

$T_{acc} = 8.4 - 3.0 = 5.4$ lb-ft
$WR^2 = 0.35$ lb-ft² (rotor only— for a typical motor)
$S = 1{,}725$ rpm

Inserting these values into Eq (8), we find that $t_{acc} = 0.36$ seconds. This calculation neglects the breakaway torque needed to overcome the static friction of the motor and conveyor bearings. However, this torque is of small duration and it becomes negligible after a few cycles of operation, during which the lubricating oil becomes more fluid. Also, the cusp in the speed-torque curve is neglected. In a well designed motor, the effect of this cusp is small. Although NEMA standards permit a drop to 70 percent of locked-rotor torque, the cusp is rarely this large.

Duty cycles that incorporate a plug-stop or a plug-reverse phase present a special problem in motor selection. This subject is discussed in the second article scheduled for the September issue of Product Engineering. Simplified solutions will be included in the article.

Knowing the equivalent horsepower during acceleration, the accelerating time, and the steady running load of the motor, we can now construct the duty cycle of the motor. From Fig. 5 we can now calculate the rms horsepower, using a modification of Eq (3):

$$HP_{rms} = \sqrt{\frac{1}{t}\left[(HP_1)^2 t_1 + (HP_2)^2 t_2 + (HP_3)^2 t_3 + \ldots . (HP_n)^2 t_n\right]} \quad (3)$$

This equation is strictly correct only when the amount of cooling is constant during the entire duty cycle. Such is the case for a self-ventilated motor running continuously at or near rated speed; or for a forced-ventilated motor (one cooled by an external system) regardless of speed. In the case under analysis, the motor is cooled by a fan mounted on its shaft so cooling effect increases with speed. Cooling during standstill and acceleration is less than cooling during the steady running period. Therefore, the formula must be adjusted to compensate for the different degrees of cooling.

$$HP_{rms} = \sqrt{\frac{(HP_1)^2 t_1 + (HP_2)^2 t_2 + (HP_3)^2 t_3}{t_{e1} + t_{e2} + t_{e3}}} \quad (9)$$

Where HP_1, HP_2, HP_3, t_1, t_2, and t_3 are the same as before, but t_{e1}, t_{e2}, and t_{e3} are equivalent durations of the three sections of the duty cycle. The equivalent duration of each period is equal to the actual duration multiplied by a constant. This constant is dependent on the degree of cooling received during that period. For self-ventilated motors running at full speed, or for forced-ventilated motors (regardless of speed) this constant is unity; i.e., equivalent time equals actual time. For small, industrial-size, self-ventilated motors, up to about 10 hp, the multiplying factor is 0.5 during acceleration, and 0.25 when the motor is at standstill. For larger motors, these factors are 0.67 and 0.33 respectively.

From previous calculations, it is known that the equivalent horsepower during acceleration is 2.76 and that during the steady running period it is one hp, maximum. For this conveyor, then, the rms horsepower (the rating required of the motor) is calculated as follows:

$$HP_{rms} = \sqrt{\frac{(2.76)^2\,0.36 + (1.0)^2\,4.64 + (0)^2\,5}{0.5\,(0.36) + 1\,(4.64) + 0.25\,(5)}}$$
$$= \sqrt{\frac{2.74 + 4.64}{6.07}}$$
$$= 1.11$$

$$\text{Ratio of } \frac{HP_{rms}}{\text{Rated HP}} = \frac{1.11}{1.0} = 1.11$$

It should be noted that the relative heating during the 0.36 sec starting period (2.74) is about 60 percent that during the 4.69 sec running period (4.64). This is a graphic illustration of how the brief accelerating period can effect the heating of the motor, even with negligible external inertia. Were the effective inertia larger, the heating during the starting period could easily exceed that during the running period.

This calculation indicates that the heating caused by the duty cycle is equal to the heating that would result if the motor were running continuously at a load of 1.11 horsepower. Had the rms horsepower been less than or equal to the motor rating (one hp) a NEMA Design-B motor would be unquestionable recommended. In this case the ratio of rms to rated horsepower is 1.11 so a NEMA Design-B motor would be subjected to the equivalent of a continuous overload of 11 percent. However, this overload is well within the service factor of 25 percent so the temperature rise will not be injurious.

An alternate for this duty cycle is the high-torque, high-slip (5 to 8 percent) NEMA Design-D motor. The Design-D motor, because of its higher accelerating torque and lower locked-rotor current (hence greater torque per ampere and per watt lost) is a more efficient accelerator. It reduces the duration and heating of the acceleration period and thereby the rms horsepower. The one hp, 1,680 rpm Design-D motor costs only 10 percent more than the 1,725 rpm Design-B motor and is built in the same frame.

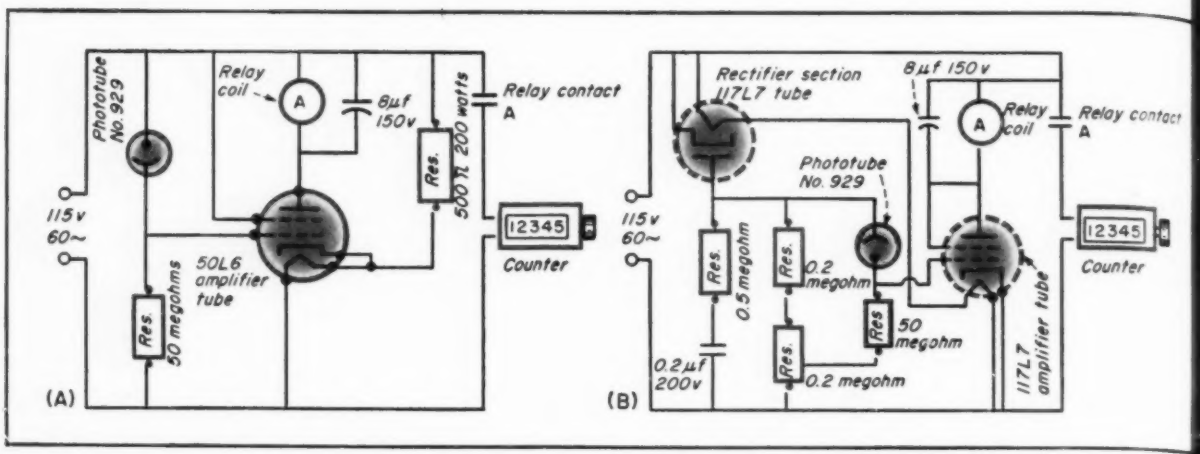
To understand the effect of a Design-D motor in reducing heating during acceleration, it must be borne in mind that the rotor I^2R loss (watt-seconds) during acceleration is identical for either type of motor. It has been proven that if friction is neglected, rotor I^2Rt loss equals the kinetic energy of the effective inertia at full speed. The principal benefit of the Design-D motor is derived from the reduction of locked-rotor current (because of the higher resistance of its rotor) and the duration of acceleration, thereby reducing stator I^2R losses and losses required to overcome external friction during acceleration. Locked-rotor current of the Design-D motor is generally about 75 percent the locked-rotor current of the Design-B, and also it decays more rapidly. The effect of the lower stator losses is to reduce the losses during acceleration about 25 percent. Since heating is proportional to equivalent horsepower squared, the equivalent horsepower of the Design-D motor is therefore:

$$HP_e = \sqrt{0.75}\,\frac{T_{avg} \times S}{5{,}252}$$
$$= \frac{T_{avg} \times S}{6{,}070} \text{ or} \quad (10)$$
$$= 0.866\,\frac{T_{avg}}{T_{rated}}\,HP_{rated} \quad (11)$$

To determine rms horsepower using the Design-D motor, the entire problem should be re-analyzed. However, the rms horsepower can be approximated from the results with the Design-B motor, as follows:

$$HP_{rms} = \sqrt{\frac{0.75\,(2.74) + 4.69}{6.07}}$$
$$HP_{rms} = \sqrt{\frac{2.05 + 4.69}{6.07}}$$
$$= 1.05$$
$$\frac{HP_{rms}}{\text{Rated } HP} = 1.05$$

Considering this small ratio, the 5 to 8 percent slip motor will be ample.



1 Phototube-relay-amplifier-counter circuits: (A) Slow acting, because a-c is applied both to the amplifier and phototube so that they both operate only part time; (B) fast acting, because d-c is used on the phototube; and (C) high speed, because d-c is applied to both the phototube and amplifier tube. The circuit at (A) utilizes only two resistors, one condenser, and a simple amplifier tube to operate the relay and phototube. That at (B), while more complex, has full line voltage on the amplifier cir-

Applying Photoelectric

WHEN COUNTING IS A REQUIREMENT, the designer usually turns first to a mechanically operated assembly, such as a lever type counter or a revolution counter because of the high reliability and low cost of such devices. If the counter is to be located remotely from the object being counted, or if more delicate operation is required, the next logical choice is an electrically operated counter, in which an electromagnet is used to move a lever arm that in turn moves a counting wheel. Operation of electrical counters, of course, depends on the opening or closing of some form of electrical switch or contact.

But if the product is light, or fragile, or small, or easily marred, or extremely fast moving, a photoelectric counting device may be applied, the reason being that only a phototube circuit can be operated without the application of a force to provide the counting impulse.

Light-beam counting is accomplished by providing a lamp on one side of a conveyor or path for the objects, and a light receiver on the other side. The light receiver is a form of electric eye relay or photo-relay connected to a counting head, which is electrically operated from the photo-relay. Normally, the light passes continuously from the lamp to the light receiver. When an object passes between the two, however, it interrupts the light beam and furnishes a count signal. After the object has passed, the light to the receiver is re-established. This provides the "reset" signal, which is equally important.

Circuit Arrangements

Several auxiliaries besides the phototube and lamp are required to complete a counting circuit: a source of power, a counter, an amplifier to amplify the signal from the phototube into a sufficient power impulse to trip the counter, and, depending on the circuit, a relay. These can be arranged in any one of several different combinations:*

1. A phototube-relay-amplifier-counter combination meets the majority of application requirements. It is fast operating and relatively inexpensive. A variety of circuits may be used with this combination, those shown in Fig. 1 being quite common and representative of the group.

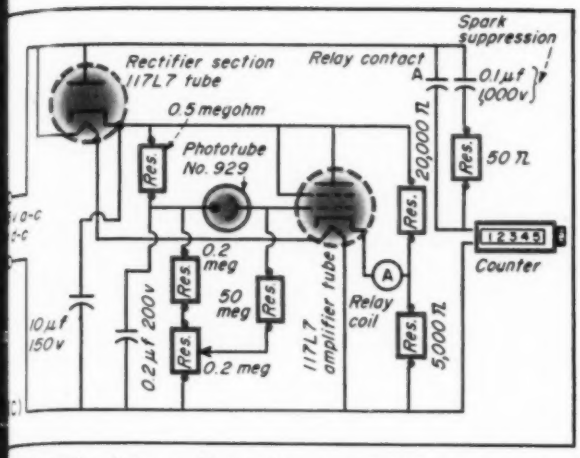
For the simplest of the circuits, Fig. 1(A), only two resistors and one condenser are used to make the phototube operate the amplifier tube and relay. In this circuit, the a-c line voltage is applied directly to the phototube and its series resistor. The phototube is operative only during part of the a-c cycle: when the cycle nears its peak, and the phototube anode is positive. During about $\frac{2}{3}$ of each cycle, it is totally inoperative, and this means that there may be a delay of about $\frac{2}{3}$ of a cycle before the phototube begins to respond. The plate circuit of the 50L6 amplifier tube in this circuit is also inoperative during the same $\frac{2}{3}$ of each cycle, and for the same reason, since only unrectified a-c is applied to the plate circuit.

When light to the phototube is interrupted, indicating the passage of an object, the current flowing through the phototube decreases in value. This reduces the voltage across the resistor that is in series with the phototube, and changes the potential of the grid of the amplifier tube to a more negative value. As a result of this, the amplifier tube no longer draws plate current. Since this plate current must pass through the coil of the relay, the relay is released, and the relay contacts close. When the relay contacts close, current passes through the electromagnet that actuates the counter, and the counter moves ahead one step.

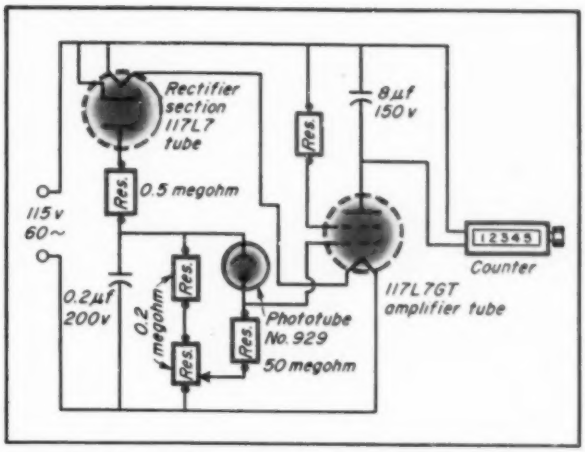
When the light to the phototube is re-established, indicating that the object has passed, the phototube current is re-established. This returns the grid potential to its initial value, re-establishes the flow of current through the amplifier tube and relay coil, opens the relay contacts, de-energizes the counter coil, and resets the counter.

The circuit shown in Fig. 1 (B) is

** Details of the design and construction of elementary electric eye relays may be found in the author's book, "Electric Eye Circuits & Relays in Theory and Practice," obtainable from Photobell Company, 116 Nassau St., New York 7, N. Y.*



cuit and a light sensitivity adjustment (which can be omitted if the light is of sufficient intensity). Besides being fast, the arrangement at (C) contains a spark suppression circuit that minimizes pitting of the relay contacts and improves operation.



2 Phototube-amplifier-counter circuit. Advantage of this type is lower first cost and maintenance cost because no relay is used. Speed of operation is about comparable to the circuit shown at (A) in the previous illustration—or slightly higher.

Counting Circuits

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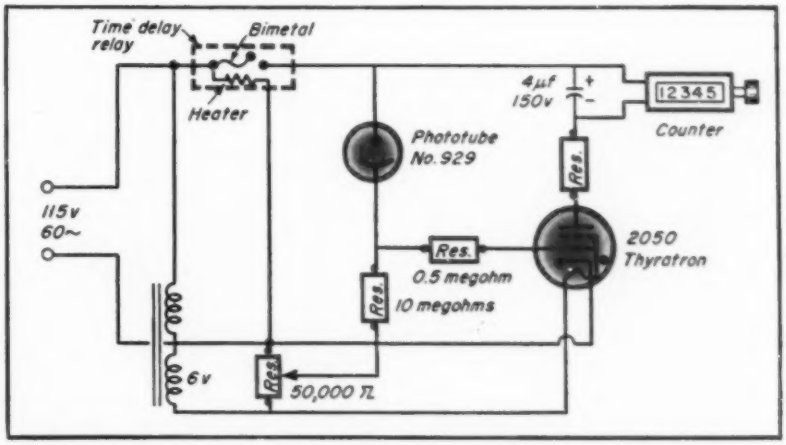
Chief Engineer, Photobell Company

a marked improvement over that in Fig. 1 (A) in several important respects. The type 117L7 tube includes a rectifier section, and this is used to rectify a small part of the a-c, furnishing d-c for the phototube so that it may operate all of the time, and respond quickly. Also, by using the rectifier in the manner shown, the d-c so obtained is negative, or lower in potential than the cathode of the amplifier tube. This is almost the same as doubling the voltage, because it allows full line voltage to be made available to the amplifier circuit.

2. Operating the counter directly from the amplifier tube in a phototube-amplifier-counter circuit, Fig. 2, has some advantages over the previous circuits. Since no relay is used, the circuit is less costly, there is one less item to service and maintain, and there is no danger of arcing or sticking contacts.

Omitting the relay also may increase the speed of operation. However, there is a disadvantage in this respect because a counting device that is operated directly from an amplifier tube usually does not receive the full rated power for which it is designed. At high speeds, the counter may fail to turn over. Therefore, the counting head must be operable on low power, as amplifier tubes do not deliver more than a few watts, and prior to final circuit approval, they should be specifically tested in the circuit under the most adverse operating conditions.

3. If the relay is to be omitted, there is an operating advantage in replacing the amplifier tube with a thyratron tube because of the larger power capacity of such a tube. In a circuit of this type, Fig. 3, a time delay relay is needed to prevent the thyratron from receiving plate voltage until the cathode has been heated; this precaution is necessary to prevent damage to the tube. Unrectified a-c is applied to the plate of the thyratron through the coil of the counter to periodically interrupt the current flow and allow the grid to regain control of the tube. (When a thyratron is conducting current, the grid cannot act as a control switch. Therefore, it is necessary for the plate voltage to decrease to zero frequently to shut off the plate current flow. At the start of each cycle of the a-c supply, the grid determines whether or not plate current should flow for the remainder of the cycle,



3 Thyratron circuit. This is similar in principle to the previous circuit except that a thyratron is used as the amplifier so as to obtain higher power outputs to operate the counter. A time delay relay is used to protect the thyratron in such a circuit, so that a time lag in operation is inherent. Also, the operating light level must be high.

and in this way fully controls plate current.)

It thus is apparent that there is a time lag in any thyratron circuit. Another limitation is that thyratrons, because of their grid current characteristics, are incapable of being controlled from phototubes that have low levels of illumination. As long as the light level is fairly strong, however, circuits of this type are satisfactory.

4. A quite different type of circuit utilizing a "photovoltaic" type of cell is shown in Fig. 4. The photovoltaic cell generates current directly from the light it receives, and furnishes sufficient current to operate a meter-type relay.

Adapting the Circuit To the Application

In adapting a photoelectric counting circuit to a practical problem or a commercial machine, certain specific problems arise that must be solved either by modification of the circuit or by selection of the counter. As indicated before, the required rate of counting is perhaps the primary factor governing the selection and modification of the circuit. For counting up to about 150 units per minute, a photo-relay and any commercial counting head usually is satisfactory. If the speed of counting must be greater, difficulties begin to appear.

One difficulty may be wearing or sparking of the relay contacts. A spark suppressing circuit, such as is shown in Fig. 1 (C), will minimize this condition. The use of large contacts on the relay also is desirable, provided the relay is not thereby required to become so large and slow that it fails to respond fast enough.

Unequal division of the time during which the current to the counter is *on* as compared to time that the current is *off* also may cause operational trouble. A typical electric counter requires an interval of current flow of about 0.05 to 0.1 sec for registering, and a similar interval of current stoppage. If either interval is reduced below this, the counter may become inaccurate. In many cases, the design of the photo-relay can be modified by the addition of a condenser-discharge delay system which increases the shorter interval.

When counting speeds approach 600 per min, only a high speed photo-relay circuit and a fast operating relay will switch the counter properly. A high speed photo-relay must employ d-c on the phototube, and also on the amplifier tube, as shown in Fig. 1(C).

To achieve speeds approaching 800 per min requires removal of the relay so that the coil of the counting device is directly operated from the amplifier tube. With this method, counting rates of up to 1,000 units per min can be obtained using the counting head shown in Fig. 5(A).

For still higher speeds, clock-face style counting heads, are practical, these being operated without relays, to reach 3,600 per min. Beyond this, electronic scaling methods can extend the rate of counting.

REVERSE MOVEMENTS OF OBJECTS. If the objects being counted are apt to move backward, as well as forward, the beam of light may be interrupted twice by the same part. When the amount of reverse movement is small enough, there is no problem. For example, assume that the beam of light is 1 in. long in the direction of movement, that the count registers when the beam of light is 80 percent interrupted, and that the count releases when the beam of light is 80 percent clear. This then would provide a "dead zone" between these two extremes amounting to 0.6 in. and reverse movements of more than 0.6 in. could provide false counts.

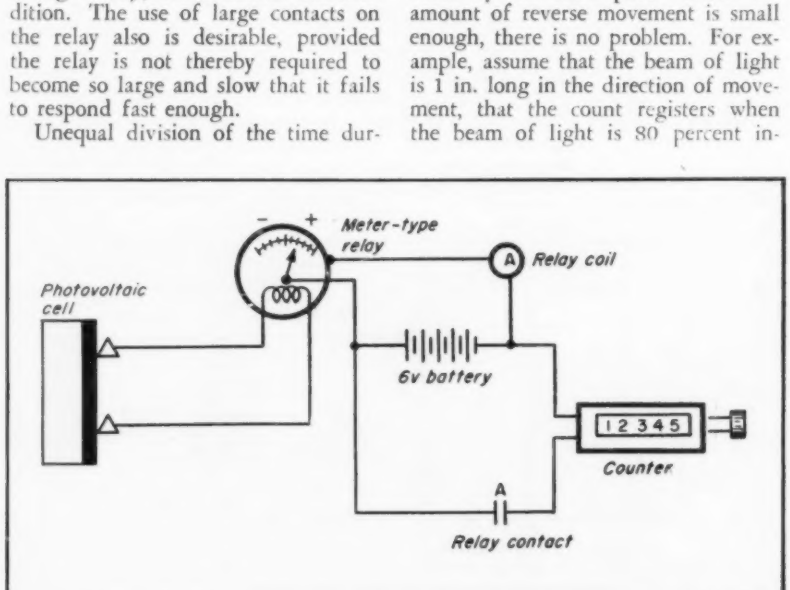
If the beam of light is made longer in the direction of movement, then more reverse movement can be tolerated. The beam of light may be broken into two parts, one from each of two light projectors side by side, and be received on two phototubes, also side by side. If the phototubes are connected in parallel, they act as two parts of the same phototube: when both are interrupted, the count registers; when both are clear, the count releases. This allows for considerably more reverse movement, and involves only the addition of an extra light projector and an extra phototube.

Protection of the counter against reverse movement also can be accomplished, often with improved performance, by using two beams of light and two phototubes, each with its own amplifier and relay, and by employing a relay circuit that performs similar functions to the above. With this design, the count occurs only when both light beams are interrupted, and an interlocking relay is tripped. The relay is released after both phototubes are clear again.

There are sometimes circumstances in which objects pass through a counting position in either direction, but must be counted with a total for one direction only, or separate totals for both directions. This can be done through the use of two beams of light, each with its own phototube, amplifier, and relay. If the light beams are interupted in the order *A, B,* then counter *AB* responds. If the light beams are interrupted in the order *B, A,* then the counter *BA* responds—or this counter may be omitted entirely if a reverse count is not required. The selection is accomplished through straightforward utilization of relays that lock up to modify the circuit for whichever result is required.

SELECTIVE COUNTING. It often is desirable to utilize some feature of an object as a counting index, so that different types or sizes of parts can be counted separately. Differentiation by color or by height are perhaps most frequently used indexes.

Two stations are required for color selective counting, each illuminated so that the light from the lamp will reflect off the part and into the phototube. If one of the parts is red and the other is blue, one phototube is made sensitive to red but not blue; the



4 Low voltage counter circuit using a photovoltaic cell. This type of circuit and counter usually is used only when battery operation is necessary. An amplifier is not required, and the entire circuit is quite simple, but its relatively high cost and slow speed of operation make it unsuitable for general purpose applications.

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other is made sensitive to blue but not red. When the red part passes, it reflects red-rich light into both phototubes, but only the red-sensitive one can respond and count. Similarly, when the blue part passes, it reflects blue-rich light back into the phototubes, but only the blue-sensitive one can respond and count. A third count of the total number of cartons may be made by use of a light beam counter that is interrupted by the passage of each part; this then serves as a check on the sum of the two color-sensitive counts.

Two stations also are required for height-selective counting. The light beams and stations are usually arranged so that one is higher than the other. Thus, the upper beam is interrupted only by the passage of the higher part; the lower beam, which is interrupted by the passage of both parts, counts the total. Another method is to set the upper light beam slightly ahead of the lower beam so that the top phototube is energized first, if at all. If energized, it sets and locks a relay between the lower phototube and the counter so that only the larger part is counted. If the high beam is not broken, the count is put on a counter for the small parts.

BATCH COUNTING. The designer often is not interested in counting, but in forming batches all having the same quantity. This is accomplished by a "predetermining type counter", as shown in Fig. 5(B). The one shown has a knob at the center of the dial, and this may be used to set the clock-face pointers to any quantity in the range from 1.0 to 400. When the actual count reaches the pre-set count, a switch responds, and this may be used for controlling a gate or any other mechanism. The design shown here may be operated up to a maximum of 500 counts per minute; it requires about one full second for resetting itself after a count. For pre-set quantities up to 16,000, two units of the same kind may be used, one controlling the other, and so providing a limit of 400^2 or 16,000.

Batching counters can also be built up from stepping relays, either of the kind commonly used in pin ball games or in telephone dialing systems. The latter are best, of course, and can be arranged to do anything in the line of batching. One type, is a newly designed relay that can count up to 2,100 parts per minute and requires no time to reset since the rachet is at the proper position for the next batch at all times.

When batching type counters require time to return to zero, it is important that no count occur during the homing interval, or else that count may be lost. By limiting the batch quantities, and by proper design, almost any desired result usually can be accomplished.

RECORDING OF COUNT. Indicating counting heads can be obtained with printing-type wheels, so that typewriter ribbon and paper can be pressed up against the wheels to furnish a printed record of the count at any given time. Also, it is possible to obtain counting heads that both indicate and print the count.

Frequently it is desirable to use a solenoid or small motor to make possible resetting of the counter by remote control. In such applications, however, it is necessary to design the circuit so that resetting is not done during the time that counting is taking place, as this may result in jamming up the counter. Similarly, an improperly-aligned count may be printed if counting occurs just as a print is being made.

A useful combination that can be created through the medium of the proper circuit, is one in which a remote pushbutton initiates a cycle during which the count coil is cut out; a print is made; the wheels are reset; and finally, the count coil is re-established into the circuit.

Insuring an Accurate Count

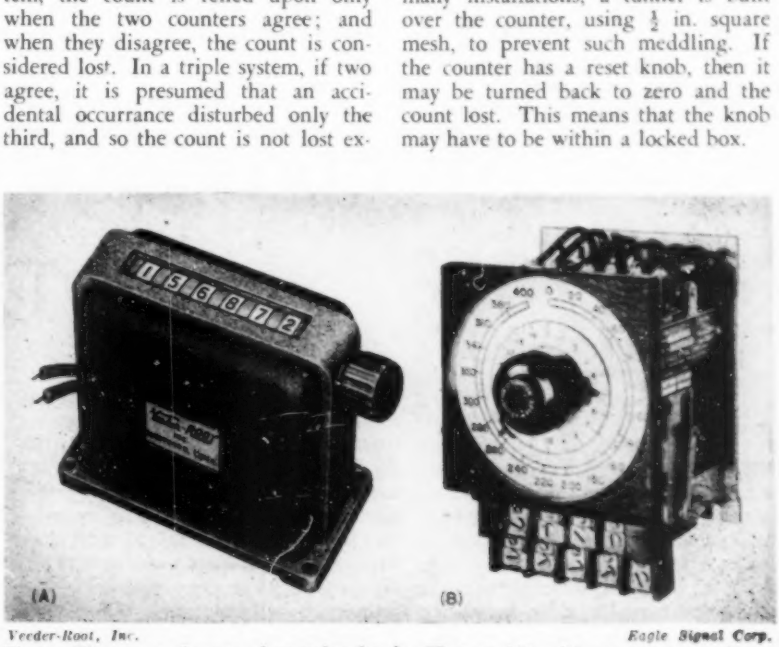
In extremely important installations, double and triple counting systems are sometimes necessary. In a double system, the count is relied upon only when the two counters agree; and when they disagree, the count is considered lost. In a triple system, if two agree, it is presumed that an accidental occurrance disturbed only the third, and so the count is not lost except in most unusual circumstances.

If the equipment on which the photoelectric counting device is mounted vibrates or oscillates rapidly, special precautions should be taken. Ordinarily phototubes are not influenced by vibration or shaking, but the light beam entering a phototube may vibrate to provide a varying amount of light, and so cause the current in the phototube to vary with the vibration. The remedy is to mount the projector and the receiver so that they both vibrate in unison, if they must vibrate at all.

If an amplifier must withstand severe vibration, then it should be equipped with shock-mounts. Also, such items as condensers inside the amplifier may have to be more solidly mounted to withstand vibration.

A relay is often sensitive to vibration, and may give double counts and spark excessively on its contacts, if not suitable. A good relay for vibration service will have a strong return spring, heavy contacts, and a wiping-type contact closure which snaps into position quickly.

When the count tends to interfere with stealing, there is a tendency to destroy confidence in the counting system. When cartons on a conveyor pass through a counting position to a shipping platform, for example, there is a strong temptation to hang a hat over the light projector, and so allow uncounted cartons to pass through. Such interference must be avoided by making the beam unreachable. In many installations, a tunnel is built over the counter, using $\frac{1}{2}$ in. square mesh, to prevent such meddling. If the counter has a reset knob, then it may be turned back to zero and the count lost. This means that the knob may have to be within a locked box.



Veeder-Root, Inc. *Eagle Signal Corp.*

Fig. 5—Two general types of counting heads. That at (A) will operate at speeds up to 1,000 counts per minute. The unit shown at (B) is a batch counter that indicates a count only after a predetermined quantity of parts has passed; it can be set to form batches ranging in quantity from 1.0 to 400, and will operate at 500 counts per minute.

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Fig. 1—Press and hammer type aluminum die forgings. Surfaces are smooth and clean because the aluminum does not scale during heating and subsequent forging.

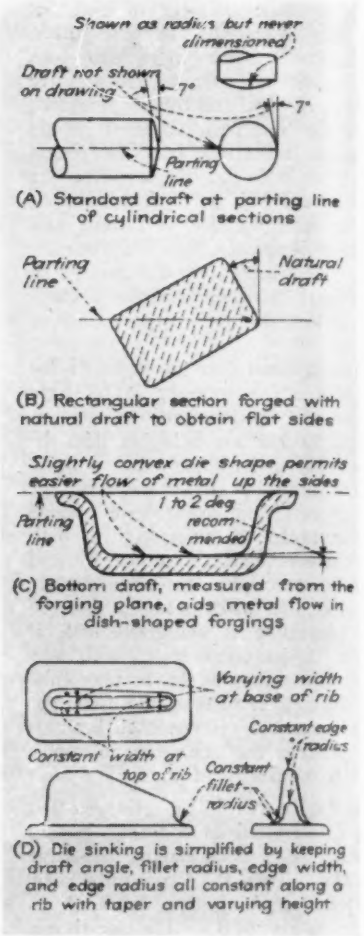
Shown as radius but never dimensioned
Draft not shown on drawing
7°
7°
Parting line
(A) Standard draft at parting line of cylindrical sections
Parting line
Natural draft
(B) Rectangular section forged with natural draft to obtain flat sides
Slightly convex die shape permits easier flow of metal up the sides
1 to 2 deg recommended
Parting line
(C) Bottom draft, measured from the forging plane, aids metal flow in dish-shaped forgings
Varying width at base of rib
Constant edge radius
Constant width at top of rib
Constant fillet radius
(D) Die sinking is simplified by keeping draft angle, fillet radius, edge width, and edge radius all constant along a rib with taper and varying height

Fig. 2—Draft requirement on various parts and how to best minimize it.

Aluminum Die Forging and Economical

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GOOD FORGING DESIGN, like design for other processes, is influenced to a considerable extent by manufacturing necessities and by the machinery employed. A knowledge of the forging process, die construction problems, and the forgeability of the alloy, will help an engineer develop sound basic designs and thereby utilize aluminum forgings to the greatest advantage.

The data presented here have been compiled from experience gained in solving numerous forging design problems. Following the rules strictly will result in better forged products at lower cost. Rigid adherence to such rules, however, is sometimes impossible. It may not be feasible to work out the functional or space requirements of the forging while complying with the proportions recommended. When this is so, the advice of a qualified forging expert should be sought.

Considerable attention to draft is warranted in design: To insure quality forgings without excessive production difficulty, and to minimize the cost of die-sinking.

Common draft angle limits are 7 or 5 deg for hammer and press forgings. Draft dimensions for 1, 5, and 7 deg draft angles are given in Table I. On mechanical upsetters, where split dies are employed, the clamping action of the gripper dies and the provision of stripper lugs or rings in their impressions permit drafts as low as ¼ deg. On presses not employing split dies, it is possible to produce forgings with

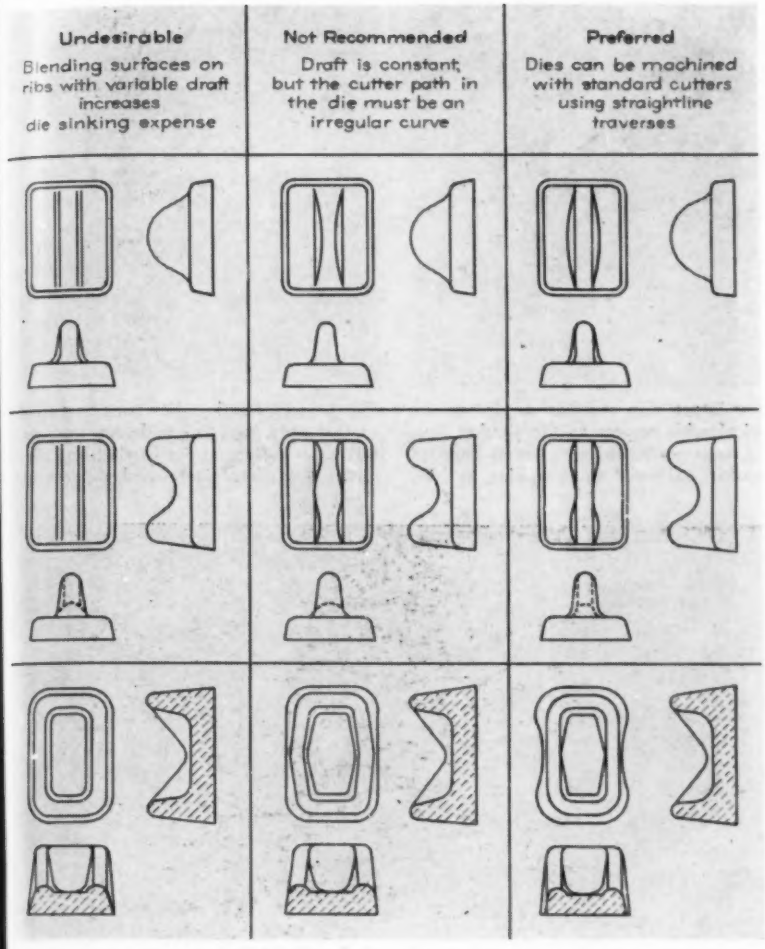
Undesirable
Blending surfaces on ribs with variable draft increases die sinking expense
Not Recommended
Draft is constant, but the cutter path in the die must be an irregular curve
Preferred
Dies can be machined with standard cutters using straightline traverses

Fig. 3—The way in which draft is specified for ribbing affects die manufacturing cost. Designs shown in the right-hand column are preferred for reasons listed.

Design for Quality Production

raft angles of less than 5 deg; but uch pieces may require a mechanical neans of aiding ejection, which would end to increase costs.

Draft measurements are expressed s angles from the axis of the hammer r press stroke, which is perpendicular o the forging plane. Even though the arting line is inclined with respett to he forging plane, or if the principal imensions of the forging are laid ut in another set of axes, draft is still eferred to the stroke of the hammer.

Natural draft occurs on the sides of cylindrical sections when they are forged as shown in Fig. 2(A). The curved surface is inherently inclined to the forging axis except at a small area near the parting line. Customary die-sinking practice is to leave narrow tangential flats in this area and maintain the standard draft angles, usually 7 deg. This draft need not be specified on the forging drawing; only a cylindrical section need be indicated.

Where draft on the sides of a rectangular section would be objectionable, it may be possible to forge the part at an angle in the die so that, as shown in Fig. 2(B), the impression will have natural draft. The resulting V-shaped cavity does not afford die life equal to that of the rectangular cavity, as the fillet in the bottom of the impression is subject to checking. This disadvantage, however, may be more than offset by economies gained at other steps in manufacture. Since the sides can be forged flat and true, an extra machining operation may be unnecessary.

Table I—Draft Dimensions For Standard Draft Angles

Depth, in.	Taper on one surface, in.		
	1 deg draft	5 deg draft	7 deg draft
1/32	0.0005	0.0027	0.0038
1/16	0.0011	0.0055	0.0077
3/32	0.0016	0.008	0.0115
1/8	0.0022	0.011	0.015
3/16	0.0033	0.016	0.023
1/4	0.0044	0.022	0.031
5/16	0.0055	0.027	0.038
3/8	0.0065	0.033	0.046
7/16	0.0076	0.038	0.054
1/2	0.0087	0.044	0.061
5/8	0.011	0.055	0.077
3/4	0.013	0.066	0.092
7/8	0.015	0.077	0.107
1	0.017	0.087	0.123

Bottom draft is desirable in wide, dish-shaped or channel forgings. The term "bottom draft" refers to a slight surface inclination measured from the forging plane, its purpose is to facilitate metal flow. This term should not be confused with the common use of the word "draft," which means an inclination measured from the forging axis to promote easy removal of the forging from the die. On the inner surface of all sections similar to that shown in Fig. 2(C), one or two degrees of bottom draft is recommended.

Die-sinking economies are gained by designing forgings with constant draft angles. Blending of two different planes for varying draft is accomplished with frequent changing of cutters, and in addition a considerable amount of hand work is often required on the die.

Ribs, sidewalls or bosses with constant draft, but tapered in depth, will vary in width at the base as a result of draft. If the cutter used in die sinking is traversed in a straight line, the top of such a rib will be straight (in the plan view) and of constant width. The base of the rib will vary in width with rib depth, and may be of curved or irregular shape. The example shown in Fig. 2(D) illustrates this principle. If it is necessary that the base of the rib be a straight

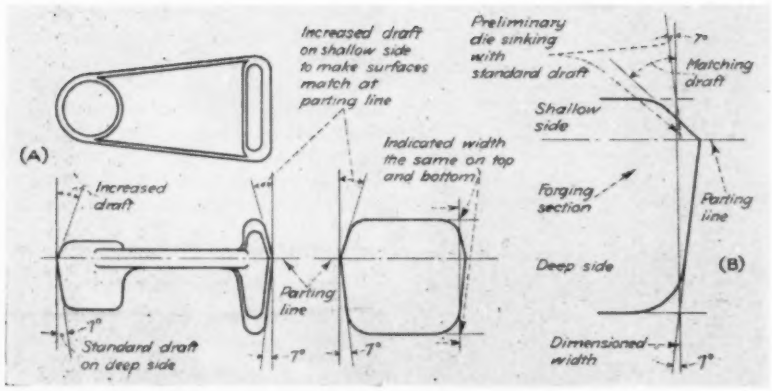

Fig. 4—(A) Method of matching draft by using larger than standard angles on the shallow side of forgings that are not symmetrical with respect to the parting line. (B) A matching draft surface that is tangent to a corner radius formed with a standard draft angle cutter. See text for details on procedure followed when sinking the die.

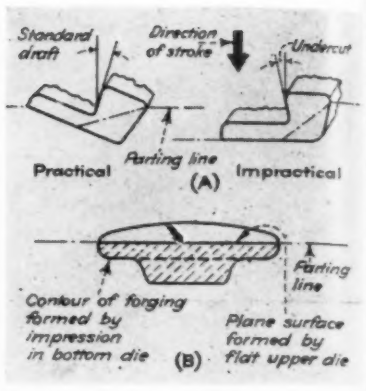

Fig. 5—(A) Forging dies must be parted so that they have no undercut surfaces. (B) a flat surface at the parting line simplifies die manufacture and die set-up.

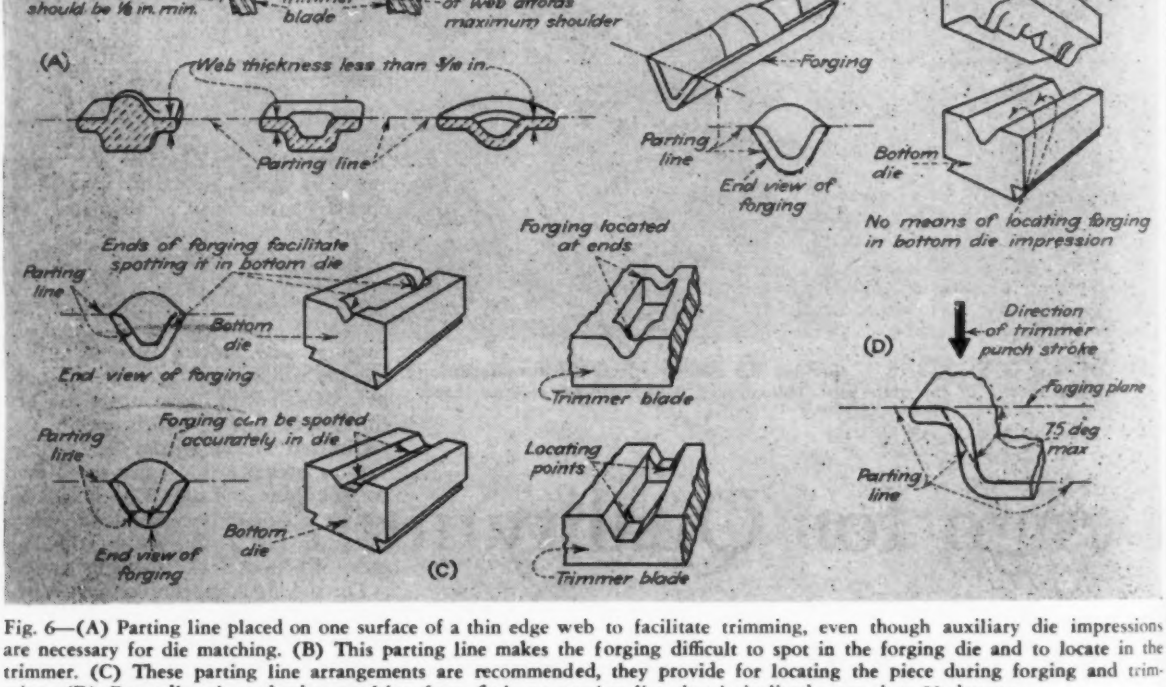

Fig. 6—(A) Parting line placed on one surface of a thin edge web to facilitate trimming, even though auxiliary die impressions are necessary for die matching. (B) This parting line makes the forging difficult to spot in the forging die and to locate in the trimmer. (C) These parting line arrangements are recommended, they provide for locating the piece during forging and trimming. (D) Raggedly trimmed edge resulting from flash on parting line that is inclined more than 75 degrees.

line or that the base be of constant width, machining the die is considerably more difficult. The traverse of the die sinking cutter must then be adjusted laterally to conform with variations in the depth of the impression. By using the shapes shown in the right-hand column of Fig. 3, tooling costs and time can be reduced.

When in doubt as to the preferred forging shape for die sinking, the designer will do well to remember that the die is a "negative" of the forging. He should visualize the various machining cutters as moving within the forged metal, forming the surfaces from the inner side. When an arduous die-sinking procedure is indicated, it should be because the use of the forging requires it, not because of a random choice in the delineation of the drawing.

Matching draft is specified when ribs or sidewalls that meet on the trimmed contour of a forging extend to different depths on opposite sides of the parting line. In die sinking, the opposing impressions are first made with standard cutters and the same draft is carried to different depths. This procedure leaves a jog or ledge at the parting line. This jog is then removed by machining the shallower impression to a greater draft angle, Fig. 4 to bring its surface out to match that of the deeper impression.

Usually, the first step in forging design is establishing the location and shape of the parting line, which may be straight or irregular, depending on the design of the forged part. This step determines the character of other

variables, such as die cost, draft requirements, grain flow and trimming procedure. Altering these factors separately and independently is possible only within narrow limits.

A number of considerations influence die parting. The die halves, obviously, must be permitted to separate, and the forging must be removable from the cavity. These limitations require that no part of the forging can be undercut. Generally the parting line, Fig. 5(A), encompasses the largest cross section of the piece, since metal can be spread easier by forging action than by forcing it into deep die impressions.

Flat-sided forgings present another opportunity to reduce die cost. By making the parting plane coincide with the flat side of the forging, it is possible to use a top die with a perfectly flat surface, Fig. 5(B). All of the impression is machined in the bottom die block. Such a design simplifies production by avoiding the possibility of mismatch between the upper and lower die impressions. It also produces forgings that are readily trimmed.

Locating the piece in the trimming die, and also in the forging die impressions should be considered when the parting line is determined. Normally, the parting line is at the center of a web that surrounds the piece. When, the web is less than 3/16 in. thick, however, the parting line should coincide with one surface of the web, Fig. 6(A). This design is desirable even though each die contains part of the impression.

With the flash at the corner of the thin web, the deepest possible ridge is presented to the trimming die for locating the piece. The advantage in trimming is gained at a sacrifice of matching edges around the web for use in checking mismatch of the forging die impressions. To check mismatch, however, auxiliary lugs can be provided for matching cavities beyond the forging impression. These lugs, if used, are sheared off with the flash in the trimmer.

Another important factor in choosing the parting line is the problem of spotting the partially forged piece in the finishing die so that it will be matched with the impression in the upper die. In the forging shown in Fig. 6(B), the parting line at the outer V-shaped surface does not provide a locating reference either in the bottom forging die or in the trimmer die blade. Although the forge shop can provide additional locating lugs or beads for spotting the piece in the forging die and for trimming, this procedure requires extra care and results are generally unsatisfactory.

A parting line at the inner surface of the V-shape or one that cuts across the bottom corner, as shown by Fig. 6(C), is more desirable. Either one of these schemes affords ample locating surfaces both for forging and trimming.

Forging of the locked-die type, on which the parting line is inclined to the forging plane, may present trouble in trimming if the inclination of the flash is too great. To prevent raggedly trimmed edges, Fig. 6(D), the flash and the parting line should not be more than 75 deg out of parallel with the forging plane.

Grain flow characteristics of the forging are affected by the location of the parting line. Excess metal flows out of the cavity into the gutter when the dies are forced together. In flowing toward the parting line, the metal may form an undesirable patern if its path is not smooth. Fig. 7(A) shows the irregularities that may result from placing the parting line at an intermediate point in a channel section. A parting line on the outer side of a rib or wall, Fig. 7(B), should follow the web inside the forging.

Choices in forged shapes often depend on parting line selection. The various treatments of the part shown in Fig. 8 are examples of detail changes that can be effected merely by changing the parting.

A simple solution is to put the large flat surface on the parting line, Fig. 8(A). This solution leaves all the impression in one die, which is an economical way to produce the part.

In Fig. 8(B) is shown a lighter version of the same part with an irregu-

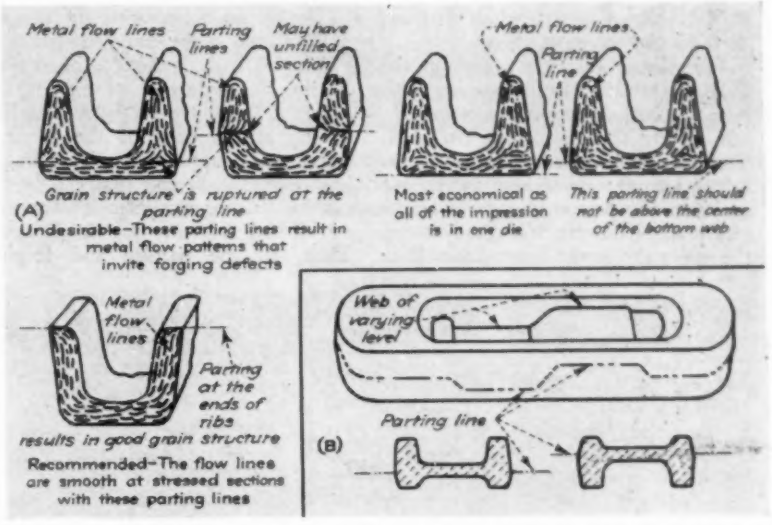
Metal flow lines
Parting lines
May have unfilled section
Grain structure is ruptured at the parting line
(A)
Undesirable—These parting lines result in metal flow patterns that invite forging defects
Metal flow lines
Parting line
Most economical as all of the impression is in one die
This parting line should not be above the center of the bottom web
Metal flow lines
Parting at the ends of ribs
results in good grain structure
Recommended—The flow lines are smooth at stressed sections with these parting lines
Web of varying level
Parting line
(B)

Fig. 7—(A) Undesirable and recommended locations of parting lines on a channel section. (B) A parting line outside a rib or wall should follow the inside web.

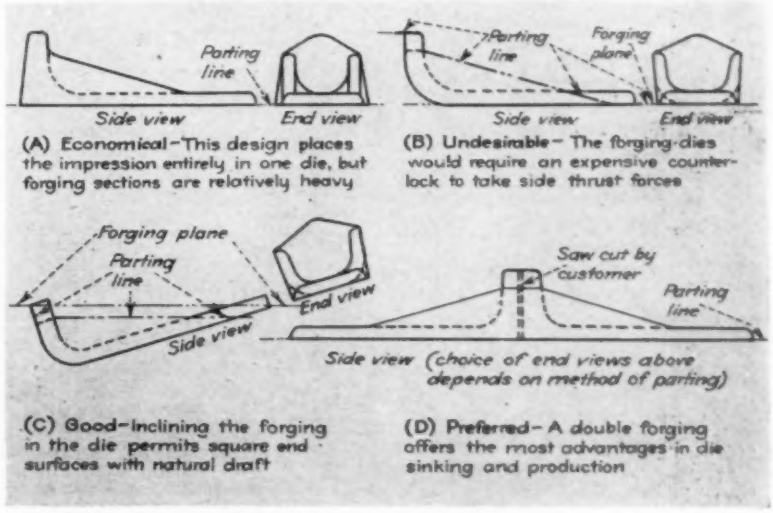
Parting line
Side view
End view
(A) Economical—This design places the impression entirely in one die, but forging sections are relatively heavy
Parting line
Forging plane
Side view
End view
(B) Undesirable—The forging dies would require an expensive counterlock to take side thrust forces
Forging plane
Parting line
End view
Side view
Saw cut by customer
Parting line
Side view (choice of end views above depends on method of parting)
(C) Good—Inclining the forging in the die permits square end surfaces with natural draft
(D) Preferred—A double forging offers the most advantages in die sinking and production

Fig. 8—Examples that show how the choice of parting line can modify a forging design.

lar parting line. It would not be desirable to forge the part in this position because of the unbalanced lateral forces exerted on the die. An expensive counterlock would have to be machined in the dies to resist side thrust.

A locked-die type of design with an irregular parting line is shown in Fig. 8(C). The forging plane, however, is inclined to minimize side thrust. This version of the part is light in weight, achieves good grain flow, and has natural draft that allows the end flange to be forged with a true right angle.

The same part produced as a double forging is shown in Fig. 8(D). Considering die cost and unit forging cost, this solution is the most economical way to produce the part. The operation of cutting the two halves apart may be advantageous when the end surfaces are to be machined.

Side thrust forces imposed on the dies as a result of forging action must be borne by ledges or a counterlock on the parting surfaces of the two dies. The press or hammer guides, in themselves, are not rigid enough and cannot maintain the precise alignment necessary to avoid mismatch errors between the upper and lower impressions. Dies with counterlocks, Fig. 9(A), are built only when the forging's shape absolutely requires them. These dies present a constant source of trouble, and they require frequent maintenance or replacement. Also, they are often considerably more expensive to manufacture than dies without counterlocks.

Avoidance of side thrust force is usually possible by inclining the forging in the die so that the forging plane and the parting line are generally parallel, that is, by placing the extremities of an irregular parting on, or very nearly on, the forging plane.

The offset connecting rod forging shown in Fig. 9(B) employs locked-die impressions because the parting line is staggered to follow the shape of the part. By inclining the forging in the die, it is possible to produce it without a counterlock

The method of locating the forging plane shown in the enlarged section of Fig. 9(B) is recommended because of its simplicity. Instead of making the forging plane intersect the parting line exactly at the ends of the forging, select points approximately ¼ in. or less beyond each end that can be located with an even dimension from an established reference on the forging. Then, let the parting and forging planes intersect at these definitely located points outside the forging.

Die-sinking practice for locked dies varies with the types of machine tools used to form the impression. Sometimes all the draft is machined with the specified angle to the forging plane. At other times, draft is first related to the parting line of the forging and is increased, only at the critical areas, in another operation. When the angle between the forging plane and the principal axis of the part is less than 3 deg, it may be practicable to make the impression normal to the parting line without any additional draft.

A double forging, Fig. 8(D), may be desirable for either or both of two reasons: (1) Side thrust on the dies and the consequent manufacturing problems associated with certain forged shapes can be avoided by producing the forgings in pairs; and (2) Right- and left-hand parts, when they are used in equal quantities, are often most economically made as pairs of opposite hand. Such a practice reduces initial die cost and saves on the price per piece.

Another article by the authors on promoting metal flow for forgings will appear in a subsequent issue.

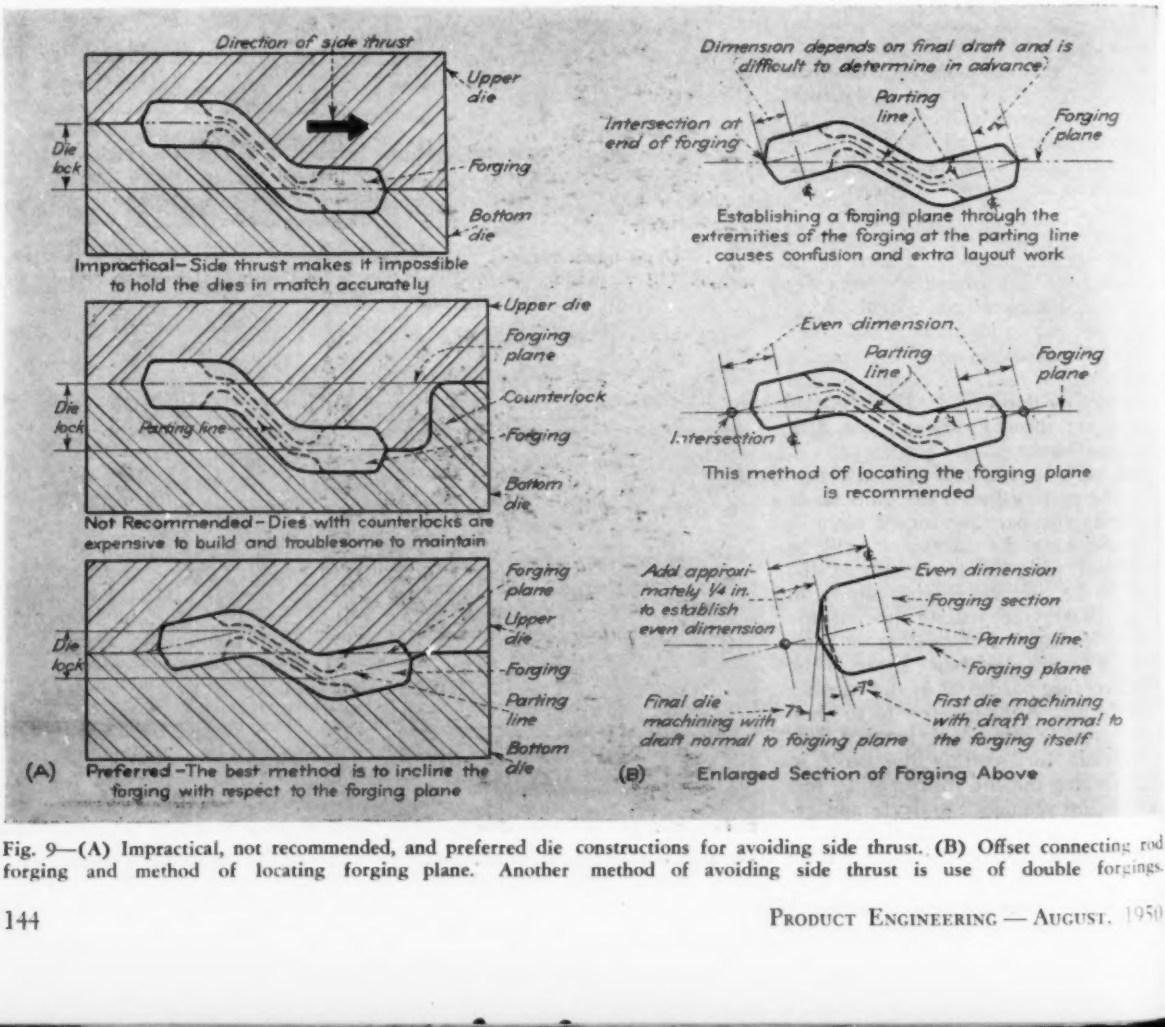
Direction of side thrust
Upper die
Die lock
Forging
Bottom die
Impractical—Side thrust makes it impossible to hold the dies in match accurately
Upper die
Forging plane
Counterlock
Die lock
Parting line
Forging
Bottom die
Not Recommended—Dies with counterlocks are expensive to build and troublesome to maintain
Forging plane
Upper die
Die lock
Forging
Parting line
Bottom die
(A) Preferred—The best method is to incline the forging with respect to the forging plane
Dimension depends on final draft and is difficult to determine in advance
Intersection at end of forging
Parting line
Forging plane
Establishing a forging plane through the extremities of the forging at the parting line causes confusion and extra layout work
Even dimension
Parting line
Forging plane
Intersection
This method of locating the forging plane is recommended
Add approximately ¼ in. to establish even dimension
Even dimension
Forging section
Parting line
Forging plane
7°
Final die machining with draft normal to forging plane
First die machining with draft normal to the forging itself
(B) Enlarged Section of Forging Above

Fig. 9—(A) Impractical, not recommended, and preferred die constructions for avoiding side thrust. (B) Offset connecting rod forging and method of locating forging plane. Another method of avoiding side thrust is use of double forgings.

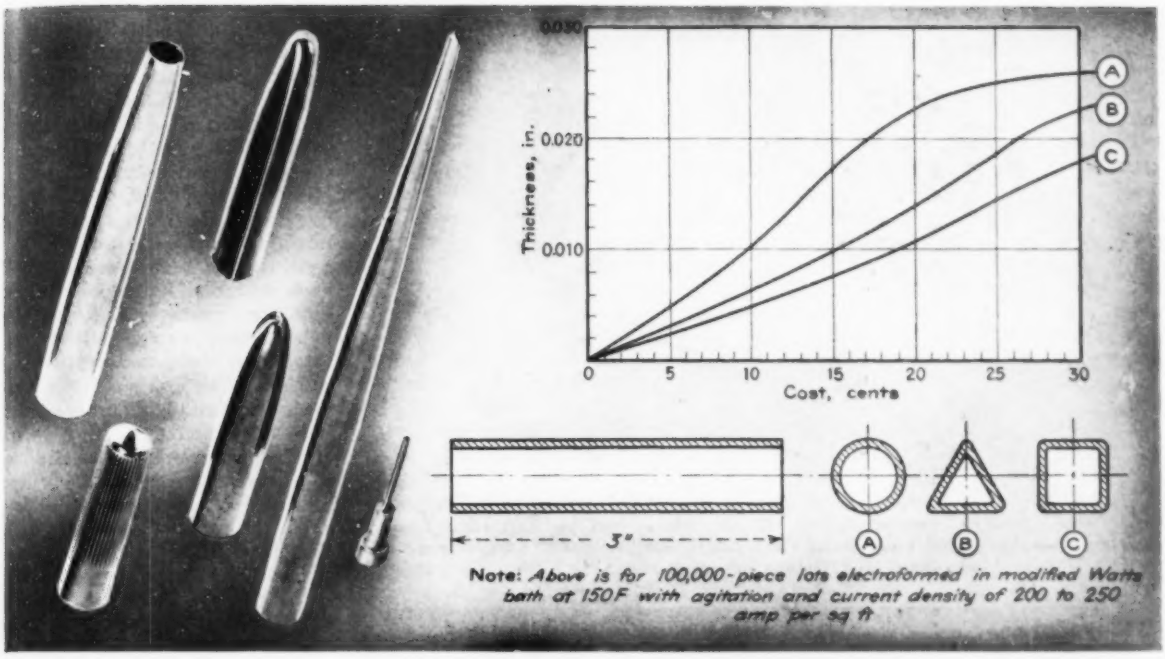

Fig. 1—Typical tubular parts electroformed in nickel on hardened-steel permanent mandrels. Cost of ribbed caps would be excessive if other fabrication methods were used. Cost of electroforming nickel at average rate of 0.010 in. per hr goes up as shape of part varies from plain cylinder. Above 0.025 in. wall thickness, advantages of electroforming become less pronounced.

ELECTROFORMING

• • • Tubular and cup-shaped parts having special contours otherwise impractical to form can now be mass-produced. Metallic coating of plastics, forming of precision dies, joining of small components, and production of limited quantities of parts for product development are other applications that are of growing interest to product designers.

RECENT IMPROVEMENTS in electroforming techniques have put this relatively old process on a mass production basis. Small tubular parts that must be produced to close tolerances and with highly polished internal surfaces are being made at a 1,000-per-hour rate. In addition, development of practical "one-shot" mandrels that are economical to make and can be re-melted after production of one part, and faster deposition of metals—particularly nickel—are other factors causing design engineers to take a closer look at the process. Here's what they are finding:

1. That many precision tubular parts can be made faster and more economically than by stamping and drawing or other forming operations that require more than a simple swedge or stamping operation (Table I). One-piece, compound-shaped tubes having any combination of bends straight edges, and twists can be electroformed with angle tolerances of ±10 min of arc. The tubular, conical and cup-shaped parts made by Camin Laboratories, as shown in Fig. 1, are examples.

2. That electrodeposition on a plastic results in a lightweight part having excellent appearance and low cost. Many types of plastics can be used as the base, and by appropriate screening, a combination of a colored plastic and polished metal surface can be obtained.

3. That the use of fusible, instead of permanent mandrels, may make

Table I—Comparison of Procedure For Fabricating a Cup-shaped Tubular Part[1]

Electroformed Nickel
1. Machine, harden and chrome plate steel mandrel
2. Electrodeposit shell
3. Polish and trim
4. Remove mandrel
Cold Drawn 9010 Bronze
1. Machine and harden blanking die and ten sets of drawing dies
2. Blank bronze disk
3. Draw—ten operations
4. Plate
5. Polish

[1] Length, 4 in.; open-end dia, ½ in.

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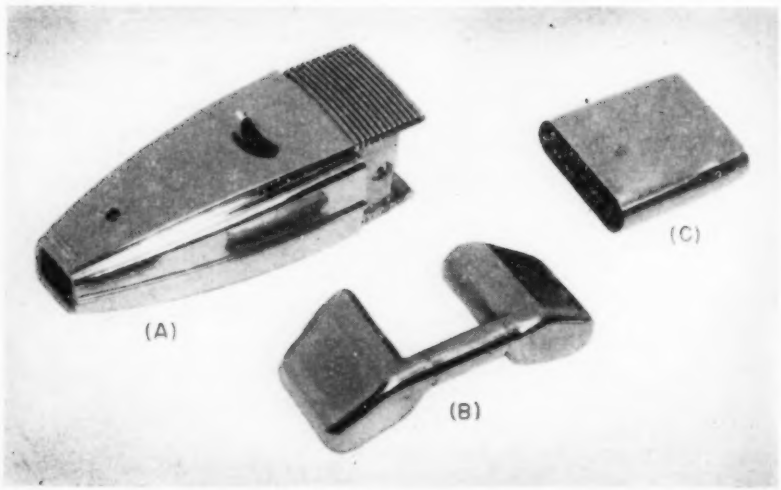


Fig. 2—Deposition on plastic electric shaver housing (A) saves weight and brings out embossed lettering. Instrument float (B) is made in two parts on fusible molds. Halves are then joined with electroformed sleeve. Simple mandrel replaces expensive dies needed to deep draw case (C) electroformed in nickel.

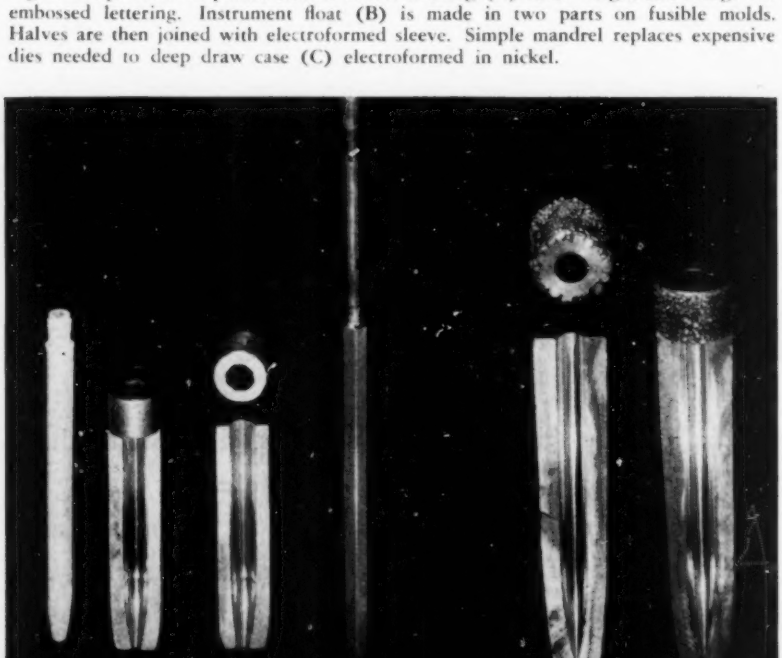


Fig. 3—Nickel mold cavities electroformed on polished steel mandrels have many applications in which plastic parts are mass-produced. Smooth inner surface resists corrosion from molding compound and lubricant, and shapes part that is free of flash.

practical a part that would be difficult or impossible to make by any other method (the mandrel being melted out of the part after it is formed, much like wax is melted out of a precision investment mold).

4. That with either permanent or fusible mandrels, limited quantities of a part can be economically produced during developmental stages for test and inspection. Even a permanent mandrel is relatively inexpensive to make since the contour is external, not internal as in a die.

5. That joining by electroforming is often less costly than by a conventional technique such as brazing or soldering.

Factors in the Process

Deposition of various metals on permanent mandrels, fusible forms or premolded plastic parts takes place in a precisely-controlled electrolytic bath. The current densities and solution temperatures used are functions of the metal being deposited and the shape of the part being formed. Length of the plated shell depends on the depth of the electrolyte tank.

Several of the metals that can be electrodeposited are applicable to this process, but the latest techniques have centered around nickel, copper and iron. In the case of nickel, the reasons for so doing are several. Compared to copper, iron and the precious metals, nickel has a relatively high deposition rate, this being 0.010 to 0.015 in. per hour when straight-section tubular parts are made. Nickel is the accepted base for chromium and other metal finishes and has a generally high hardness range, dependent on type of bath and rate of deposition, as well as the shape and finish required for the part.

Exact reproduction to mandrel dimensions permits mass duplication of parts to close tolerances. With permanent, hardened steel mandrels that are machined and precision ground, ID tolerances of ± 0.0005 in. can be attained. Thickness is, of course, a function of the plating time and current density. Plating costs (Fig. 1) are, therefore, proportional to the thickness required, the complexity of the shape and the number of parts to be made.

Close tolerances can also be held on complicated shapes incorporating undercuts and special contours. The fusible mandrels required for such parts must be carefully cast and finished. In this case, tolerances are usually held to ± 0.002 in. Greater accuracy is possible, but the economics of reproducing the "one-shot" disposable mandrels make smaller tolerances impractical.

Special assemblies like the instrument float shown in Fig. 2 illustrate the use of fusible mandrels and the application of electroforming as a means of joining the components of an assembly. The left and right bulbs are plated on fusible forms. After the latter are melted out, the halves are jigged into position and the necks "soldered" together with a "plated sleeve".

Capillary tubing having a minimum wall thickness of 0.005 in. and an ID of 0.002 in. can be formed on a wire mandrel. The difficulty and high cost of drawing and annealing such small tubing is thus avoided. By the same token, precision tubing having nonstandard dimensions is easily produced.

Mold Cavities

Electroformed cavities are gaining popularity as substitutes for steel molds used in the mass-production of long slender plastic parts like the hexagon-shaped pencil barrel, Fig. 3, and aircraft nose intake ducts, Fig. 4. Mass-production runs of small parts are made using a series of these cavi-

ies electroformed to close tolerances on the master mandrel. The cost of these cavities is substantially lower than that for hobbed molds.

The advantages in this molding application are several. When the part emerges from the cavity, there is no flash and a smooth finish. Simple trimming is usually the only secondary operation required. The corrosion resistance of the nickel mold contributes to long-run economy, because the chemicals in both the molding compound and lubricant are counteracted. Atmospheric corrosion during storage between production runs is also a factor that simplifies overall manufacturing procedures.

Electrodeposition on a Nonmetallic Base

Decorative and electrical coatings on plastics and non-conductors represent still another field for electro-deposited metals. Many of the thermoplastic, thermosetting or non-plasticized synthetics can be plated. These materials are made conductive with a film of pure silver. Nickel or copper is then electroformed on the silver, followed by flash or heavy coatings of chromium, gold or other metals. Final plating can be relatively thin if a decorative finish is desired, or thicker if a wear-resistant or stiffening case is required.

Economical sandwich constructions are also possible. Materials costs can be reduced by plating a plastic core with copper if an electrically or thermally conductive surface is required. Overall weight of parts can be decreased with a laminated construction of this type.

Electropolishing

Polishing the external surfaces of electroformed parts is a second operation necessitated by the dull finish that characterizes as-plated pieces. To solve this major problem and to eliminate buffing, a process of deplating is being perfected. Applicable to all plated metals, this process has several advantages when compared to average buffing procedures. Less metal is removed, so that existing tolerances can be preserved. There is no danger of work-hardening the plate, as is sometimes caused by heat generated during buffing operations; and subsequent tarnishing is retarded through passivation of the surface.

Design Tips

As previously mentioned, thick wall sections require extended plating periods and increase cost. Therefore, the designer should specify uniform wall thicknesses as thin as are consistent with functional requirements of the part.

Generous fillets should be used to increase strength and rigidity in complex parts and to simplify plating procedures. If true right angles are required as in rectangular-section copper wave guides, special attention should be given to current screens, shields and wiring to prevent under-or over-buildup at the vertex of an angle.

Dimensions of recesses and grooves are somewhat critical if complicated plating procedures are to be avoided. A good general rule is to specify a groove length that is two to three times the depth.

On tubular parts with a closed end, a slight taper is needed to permit separation of the shell and mandrel. If no taper can be tolerated along an appreciable length, a pin hole through the closed end will prevent formation of a vacuum.

The precision model of a complex shape should be designed so that the reverse matrix on which the mold is electroformed can be easily removed.

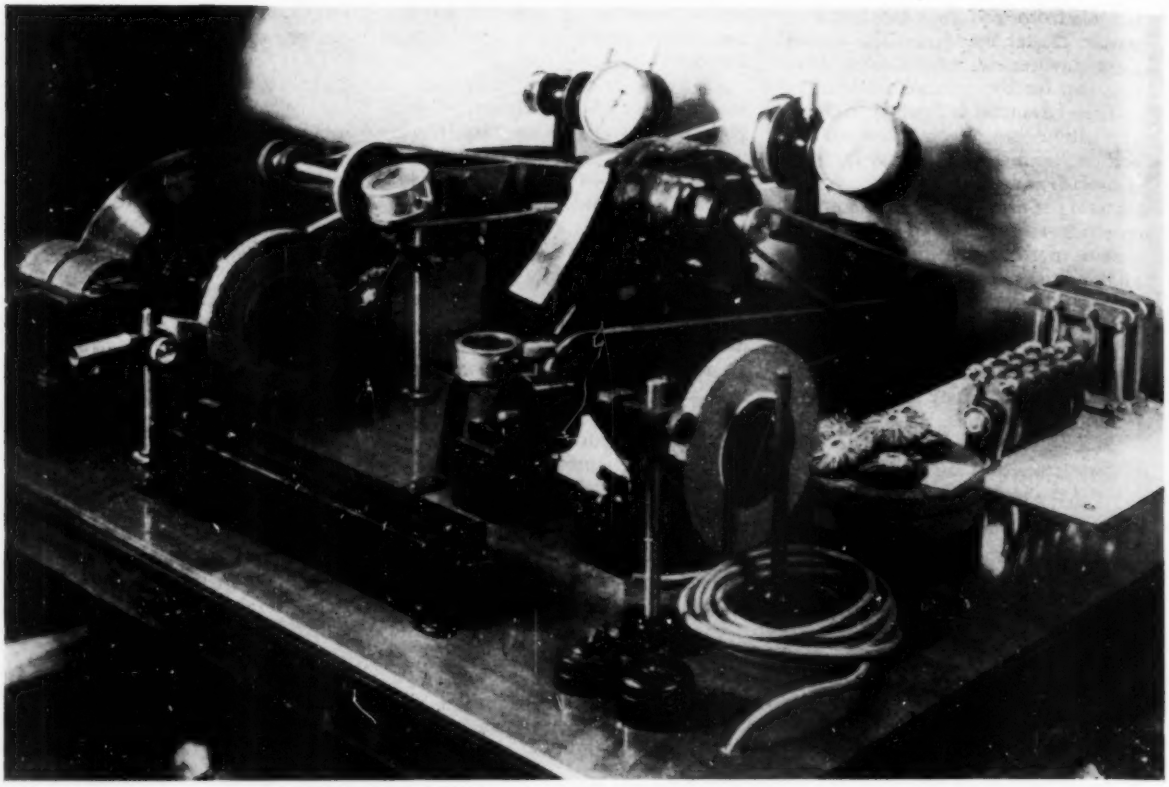
Parts requiring internal threads and other screw-machine operations should not be electroformed. The time required is many times that of most screw machine cycles.

In high-pressure applications such as dies for injection molding of plastics, parts are usually limited to shapes for which the outside surface of the electroformed cavity can be machined to fit into a recessed die block. It is also possible to build up the contour of the outside surface with sprayed metal so that the entire surface can be machined.

EDITOR'S NOTE: Special credit is due the Camin Laboratories Inc. of Brooklyn, N. Y. from whom the majority of the material for this article was obtained. Acknowledgement is also made to C. L. Duncan of White Plains, N. Y.; International Nickel Co., Inc.; Bone Engineering Corp., Eberhard-Faber Pencil Co. and F. A. Ringler.

Bone Engineering Corp.

Fig. 4—Mold for aircraft nose intake duct electroformed in nickel. Design calls for heavy wall section with pressure plate working at 50 to 100 pounds per square inch



THE LAMBOURNE ABRADER gives controlled laboratory results which correlate very well with actual road tests on tires.

RUBBER *for abrasion resistance*

"Abrasion resistance" of rubber is variable, depending on which one of many testers you use. This property must be related to other requirements. The designer has a choice of natural and "chemical" rubbers; different compounding methods and ingredients make available many types.

CHARLES W. WAHL
Hewitt Rubber Div., Hewitt-Robins Inc.

ABRASION IS OFTEN CONSIDERED as the resultant of several properties—including tearing and cutting—not as a specific property in itself. This is probably truer of rubber than of rigid materials such as metals. In actual service, abrasion resistance of rubber products cannot be considered alone, but the concomitant properties of tensile and compressive strengths, hardness and resilience must be considered at the same time.

It is for this reason that no laboratory machine has ever been devised which can evaluate the abrasion resistance of rubber for all types of service. However, certain machines are claimed to give results that correlate well with specific types of service. For example, the Lambourne Abrader is said to give results closely related to tire tread wear, and the Sproul-Evans Abrader to sandblast hose tube service life. Of the following seven types of abrasion testers, the first five are based on standard ASTM methods, and the last two are non-standard and quite variable in method.

Table I—Comparison of Tests Made on Two Different Abraders
(based on compound No. 1, the best, taken as 100.)

Compound No.	Lambourne Abrader	U. S. Rubber Abrader
1	100	100
2	79	91
3	66	98
4	50	87
5	49	91
6	44	90
7	33	58
8	32	73
9	18	48
10	10	36

NATIONAL BUREAU OF STANDARDS ABRADER. 1 x 1 x $\frac{1}{4}$ in. specimens are held by a downward force of 5 lb against the periphery of an abrasive-

covered drum rotating at 40 rpm until 0.1 in. of rubber has been abraded from the samples.

THE DU PONT ABRADER. 2 x 2 x 1 cm specimens are held by a 3.62 kg weight against the flat surface of an abrasive disk rotating in a vertical plane at 37 rpm for 20 min.

NEW JERSEY ZINC COMPANY ABRADER. 2 x 2 x $\frac{1}{4}$ in. specimens of rubber are held against the flat surface of a horizontal abrasion track 30 in. OD and 24 in. ID revolving at 30 rpm for 1,000 revolutions. The specimens can be alternately raised and lowered to give tearing and cutting action, in one modification of this machine.

UNITED STATES RUBBER COMPANY ABRADER. 2 x 3 x $\frac{3}{8}$ in. specimens of rubber are held in a specimen carrier, which moves the specimens against an abrasive wheel rotating against the direction of movement of the specimen. The rotation of the wheel is reversed at the time the movement of the specimen carrier is reversed. A test is run until a given number of cycles have been completed.

KELLY-SPRINGFIELD ABRADER. $2\frac{1}{2}$ x $5\frac{3}{16}$ x $\frac{1}{4}$ in. specimens are mounted tangentially by one end to the periphery of a 10 in. wheel made to revolve at 70 rpm. A 6 in. abrasive wheel travels against the face of each specimen at the rate of 2,386 fpm. A test is run for 675 revolutions of the specimen.

THE LAMBOURNE ABRADER. A molded rubber wheel is rotated so that its periphery contacts the flat surface of an abrasive wheel. In variations of this test, the specimen can drive the abrasion wheel, or vice versa, or both can be driven.

THE SPROUL-EVANS ABRADER. The specimen is rotated while completely immersed in abrasive particles.

Most methods of determining abrasion resistance have these points in common: After a definite "break in" period, the rubber specimen is removed from the machine and weighed. After testing, it is weighed again and the volume lost is calculated. This may be compared with a standard, the performance of which is known.

Although such deteriorating factors as overvulcanization and aging will generally lower the abrasion resistance, both in service and in testing, not all formulations can be tested. For instance, compounds which "gum up" the abrasive give false results.

The first six machines are of the same "type" and yet results among them do not correlate. Table I shows results of tests run on two types of abrasion machines. In general, a tire tread type compound gives superior results on this type of machine and results are roughly like those obtained in service with a tire tread, belt cover, or chute lining. The seventh machine listed seems to be in a class by itself, showing superior results with a highly resilient type of rubber which is best for sandblast service.

Table II—Types of Rubber Used in Abrasive Service

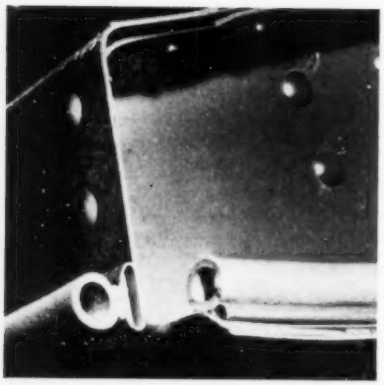
Trade or Commercial Name	Chemical Type	Sp. Gr.	Useful Properties	Typical Applications
Natural Rubber	Isoprene polymer	0.92	Very good abrasion resistance High resilience and good tear	Belt cover Sandblast hose tube
GR-S-50 (general purpose "synthetic" rubber)	Butadiene-23½% Styrene copolymer	0.93	Good heat resistance	Hot materials belt cover
GR-SX485	Butadiene-29% Styrene copolymer	0.93	Very good abrasion resistance	Tire tread
Butyl GR-I-50	Isobutylene-2% Isoprene copolymer	0.91	Good impermeability to gases Good acid resistance Good aging in air Good aging in steam	Tire inner tube Acid hose tube Hot materials belt Steam hose tube
Neoprene GN	Chloroprene polymer	1.25	Good sunlight resistance Good flame resistance Good metal adhesion Fair oil resistance	Garden hose cover Welding hose cover Roll cover Fuel oil hose tube
Buna N	Butadiene-26% Acrylonitrile copolymer	0.97	Good oil resistance	Gasoline hose tube
Buna N	Butadiene-45% Acrylonitrile	1.00	Very good oil resistance	Gaskets for oil service
Thiokol FA	Sodium Tetrasulfide-Ethylene Dichloride Copolymer	1.34	Excellent sunlight and solvent resistance	Paint spray and curb pump hose tubes and covers
Polyacrylic rubber (Hycar PA)	Acrylic Acid Ester polymer	1.1	Excellent heat resistance	Gaskets for hot service
Silicone Rubber	Silicon-Oxygen copolymer	1.2	Excellent cold and heat resistance	Gaskets, tubing, wire jacket for extreme temperatures

Table III—Properties of High Gum Stocks

	Tensile Strength psi	Elongation percent	Shore Durometer Hardness	Resilience percent	Relative Tear
Natural Rubber	3,000	800	35	70	100
GRS-50	600	550	25	50	25
Butyl (GR-I-50)	2,500	800	30	15	25
Buna N (26% Acrylonitrile)	700	500	45	55	35
Buna N (45% Acrylonitrile)	1,000	600	45	25	50
Neoprene (GN)	3,500	800	45	60	65
Thiokol (FA)	165	310	41	50	15

Types of Rubber

The unique properties of the new "synthetic" or chemical rubbers assure them a place in engineering despite higher prices or more difficult process-



SILICONE RUBBER (Silastic 250) is extremely good for specialty applications where abrasion resistance is required over a wide (—100 to 160 F) temperature range.

SANDBLAST HOSE in cross-section shows the soft resilient tube which takes harsh abrasion of cutting sand and shot. Tube extra thick for long service wear.

Table IV—Properties of Tread Type Compounds

	Tensile Strength psi	Elongation percent	Shore Durometer Hardness	Resilience percent	Relative Tear	Order of Best (1) to Worst (7) du Pont Mch. Abrasion Resistance
Natural Rubber	4,200	575	63	50	100	5
GR-S-50	2,500	550	60	40	30	4
Butyl (GR-I-50)	2,400	600	60	10	70	7
Buna N (26% Acrylonitrile)	3,500	450	68	40	35	2
Buna N (45% Acrylonitrile)	4,000	450	75	20	40	1
Neoprene (GN)	3,400	500	65	45	50	3
Thiokol (FA)	1,200	450	71	25	20	6

ing of some types. Table II lists representative types of chemical rubbers and natural rubber used in abrasive service, together with some of their unique properties and suggested applications.

It is frequently necessary to sacrifice one or more properties to get the desirable properties which suit a rubber compound for a particular service. In any single compound, the best abrasion-resisting formulation may not be obtained because of these limitations:

1. Cost: High reclaim, high load, or cheap fillers may reduce cost but also reduce abrasion resistance.

2. Color: Maximum abrasion resistance is obtained by using carbon blacks; however, new light-colored pigments—fine particle-size silicas—show good abrasion resistance in light colored stocks.

3. Specific properties obtained by compounding: Low durometer (very soft) stocks, maximum heat resistance or high dielectric strength compounds are examples of compounds which are likely to be poorer in abrasion resistance.

Outside of the polyacrylic and silicone rubbers, which should be considered for specialty uses, the remaining rubbers can be listed as follows, in order of decreasing abrasion resistance:

GR-S X-485	Very Good
Buna N	Very Good
Neoprene Gn	Very Good
GR-S-50	Very Good
Natural Rubber	Very Good
Butyl	Fair
Thiokol FA	Fair

The X-485 GRS is the popularly termed "cold rubber". Actual road tests have shown that cold rubber has about 20 percent more life than natural rubber treads. However, not all X485 GR-S compounds, for example, will be better abrading than natural rubber compounds; it depends on the compounding. Therefore, the foregoing listing is only for one type of compound which is supposed to exhibit the best abrasion resistance to the exclusion of other, possibly just as important, properties.

General Properties

The chemical rubbers are more difficult to vulcanize than natural rubber but are generally more resistant to aging, chemicals, and other agents which tend to destroy them. Polyacrylic (Hycar PA) and silicone rubber are specialty rubbers which, because of their high price, have very limited service. However, in certain applications their cost is well justified. Hycar PA can be used at a temperature of 300 F while silicone rubber can be used in a —100 to 500 F temperature range.

Table III shows the relative properties of natural rubber and the commonly used chemical rubbers in "high gum" stocks—those with little or no filler and oil, and containing only the necessary vulcanizing, anti-oxidant and sun-resistant agents. Table IV shows the relative properties of natural rubber and the chemical rubbers in "tread type" stocks which are loaded with about 50 parts of carbon black.

The ability of some of these rubbers to withstand heat aging is shown in Table V. Deterioration of rubbers at room temperature over a period of years is somewhat similar to deterioration of them at elevated temperatures over a period of days. Butyl is a good low cost rubber for service at elevated temperatures (250 F).

Table VI illustrates the difference in volume increase of the various polymers in regular gasoline and in a 40 percent aromatic fuel. These tests were made on high gum stocks. Compounds containing oil and filler will swell considerably less. This table clearly demonstrates, however, the superiority of some of the chemical rubbers to natural rubber in resistance to this type of deterioration. High volume increase in a rubber compound produces a "tender" compound with reduced tensile, tear and abrasion properties.

Abrasive Applications of Rubber

Where rubber can be used to line steel, such as in ball mills, chutes, concrete mixers, and similar extremely

Table V—Heat Aging Characteristics of Natural and Chemical Rubbers

(Oven Aging at 250 F)

	Percent of Original Tensile	Percent of Original Elongation	Durometer Hardness Change
Natural Rubber	20	35	+6
GR-S-50	85	45	+16
Buna N (26% Acrylonitrile)	85	35	+16
Neoprene GN	40	25	+20
Butyl GR-I-50	90	85	+4

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Table VI—Resistance to Fuels

(Percent volume increase of high gum stocks after 72 hr immersion at room temperature)

	Gasoline	40 percent Aromatic Fuel
GR-S-50	300	500
Natural Rubber	290	385
Butyl GR-I-50	265	280
Neoprene GN	50	185
Buna N (26% Acrylonitrile)	25	90
Buna N (45% Acrylonitrile)	3	45
Thiokol FA	2	15

abrasive services, a direct comparison between steel and rubber has been made. Rubber liners have been reported to outlast steel liners by ratios from 5:1 to as much as 25:1. In addition, rubber resists corrosion by both acidic and basic chemicals.

Although most applications of rubber and rubberlike materials require some abrasion resistance, many are not thought of specifically for this requirement. However, to understand the principles of applying rubber in abrasive services, two types of products requiring a high degree of abrasion resistance—hose and conveyor belts—are described here. It is hoped these principles can be carried over to other rubber applications.

HOSE. Hose designed to carry abrasive materials can be classified into two types: (1) Those carrying liquids containing large amounts of abrasive material, such as sand suction hose and dredge sleeves; and (2) those carrying dry grit, such as sandblast hose. The latter type has a highly resilient tube which contains little or no "loading" ingredients. Where the buildup of electrostatic charges is likely to occur, many users of sandblast hose prefer a tube compounded with electrically conductive carbon black. Sand suction hose tubes and dredge sleeve tubes, however, are compounded with greater amounts of carbon black to give maximum abrasion resistance for this type of service.

Abrasives-carrying hose are constructed with extra thick tubes, $\frac{1}{4}$ to $\frac{1}{2}$ in. wall thickness, to obtain maximum service life. The inner surface must be very smooth, since even very small projections will cause turbulence and consequent wear on one area. To prolong the life of these types of hose, the designer should make provision in the equipment for: (1) Keeping the hose as straight as possible during use to maintain minimum impingment of the abrasive against the hose tube; and (2) if the hose must be bent, rotating it at intervals to prevent continuous impingment at one point.

Hose covers are compounded to give maximum abrasion and tear resistance, especially for hose that is dragged over sharp stones or rocks. Cover compounds are compounded also with sun resistors to minimize "checking" caused by sunlight.

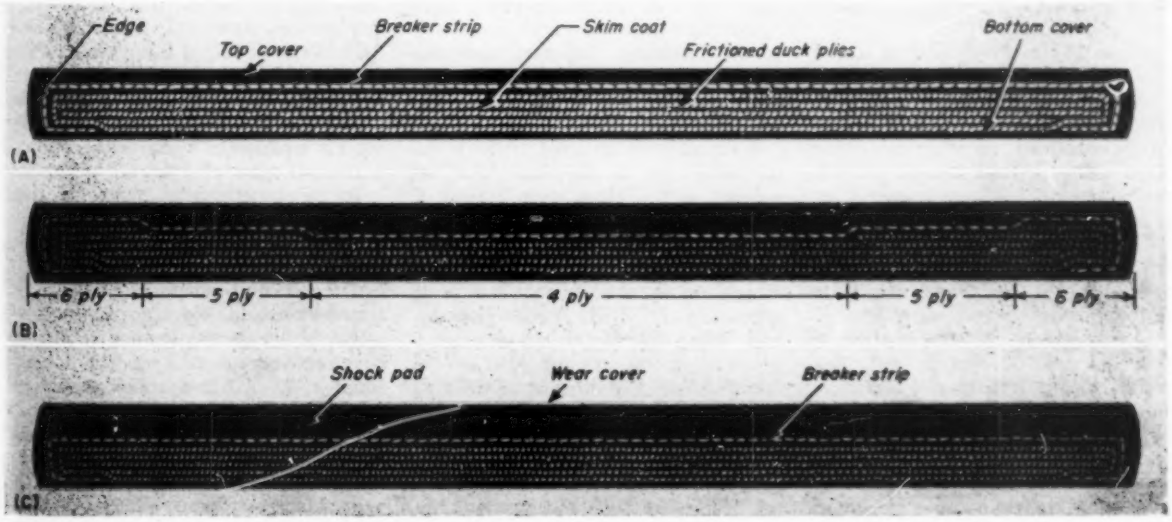
CONVEYOR BELTS. Like hose, rubber covered conveyor belts are used to carry a wide variety of materials—everything from foundry sand to granite and flint rock. Cover thicknesses for the belts vary with the material carried, size of the lump, and speed of the belt. Since water is a lubricant in cutting rubber, lumps with sharp edges are much more harmful to a rubber cover when wet. Chemical rubber covers need to be about one-third thicker, generally, than natural rubber covers.

Conveyor belts are constructed straight-ply or stepped-ply. The stepped ply belt has fewer plies at the center of the belt, allowing for a heavier cover, which absorbs shock better and lasts longer before wearing through. The stepped-ply belt, having fewer center plies, "troughs" more easily and conforms well to the idlers. A shock-pad construction is also available for severe service. This pad of highly resilient rubber helps to protect the cover from puncture and also shields the duck from rupture by yielding to sudden shock loads.

Cord construction belts are superior to fabric construction belts in ease of troughing, and in ability to withstand impact, tearing, cutting and possibly abrasion.

Quality of belt covers is comparable to that of tire treads, and covers are available in three or four grades of rubber for cold non-oily materials. The best grade available will generally contain no reclaimed rubber and will withstand severe abrasion, tearing and cutting. The lower grades are less expensive and are designed for lighter, less abrasive services.

Neoprene or Buna N belts are made for carrying oily materials such as oil treated coal. Neoprene, however,



THREE TYPES OF BELT CONSTRUCTION: (A) The straight ply belt is the general purpose type. (B) The stepped ply belt has more rubber covering in the center for longer wear. (C) The shock pad belt has a resilient cushion under the top cover to absorb the sudden shock of heavy loads and give longer life to the wearing surface of the belt. It also reduces shock to plies.

Table VII—Hot Materials Belts

Condition of use	Maximum Temp., F	
	Cotton Fabric Belt	Synthetic Fabric Belt
When belt is running in a closed chamber	200	200
When belt is carrying fine material	250	250
When belt is carrying lump material	275*	350

*Cotton fabric chars at about 300F

has the disadvantage of stiffening in cold weather and therefore its use as a belt cover is limited to temperatures above 0 F. Plasticizers which would help neoprene to retain its flexibility at lower temperatures rob it of much of its abrasion resistance.

Use of a natural rubber or GR-S belt in even mild or intermittent oil service is not recommended because the swelling which may take place in the top cover of the belt will give the belt a contour opposite to that of the idlers and not allow it to run properly.

A belt cover not compounded for heat service will have greatly reduced life if used at temperatures higher than about 150 F. Above this temperature, a special hot materials belt should be used; one of the chemical rubbers is used for this type of belt. GR-S or butyl can be used for non-oily hot service, but neoprene or Buna N should be used for hot oily materials even though their cost is greater. Table VII specifies maximum temperatures recommended for three distinct kinds of hot material service. With one exception, these temperatures are based upon the ability of the cover compound to withstand the temperature.

Much can be done in the design of a conveyor belt system to minimize cover wear and fabric damage. The critical point of a conveyor is the point at which it is loaded, for here it receives its greatest abrasion and nearly all its impact. Material should leave the chute at the same speed and direction of travel as the belt and land on it with the least amount of impact. Fine material should fall on the belt first to provide a cushion for the heavier material. This can be accomplished by use of a slightly fanned row of bars at the bottom of the chute. The material should not hit the belt over a pulley. However, if this is unavoidable, the pulley should be rubber covered or of the pneumatic type.

To prevent wedging of material between chute and belt, there should be a minimum of two inches between the steel chute and the belt, and this distance should be made increasing in the direction of travel of the belt. The belt at the loading point should be in a troughed position. The belt should be designed to run at a speed such that the belt will be filled to capacity. This allows for minimum belt speed and consequently minimum belt wear.

It should be remembered that few applications of rubber demand maximum abrasion resistance; this allows the various rubbers to be compounded in such a way as to obtain an optimum balance between cost and performance.

Spring Reverses Electric Motor in Four Milliseconds

A METHOD FOR REVERSING a small electric motor in three to four milliseconds has been developed at the National Bureau of Standards by Jacob Rabinow, inventor of the much-publicized magnetic clutch. Although intended specifically for attaining high-speed reversal of magnetic tapes in electronic digital computing machines, the technique may prove useful in many other applications.

While it is relatively easy to stop a motor quickly with brakes, rapid starts in either the same or the reverse direction are limited by the low torques that can be obtained by electromagnetic means alone. In the Bureau's rapid-reversal motor, the kinetic energy of the rotor, instead of being dissipated as heat in a brake during deceleration, is converted into potential energy in a spring, which is then used to accelerate the rotor in the opposite direction.

A small low-inertia two-phase motor operating at 3,200 rpm is used. The reversal spring consists of a steel torsion bar approximately 31 in. long and $\frac{3}{16}$-in. in diameter. Only one phase of the motor is connected to the a-c power supply; thus the motor will rotate in the starting direction, either clockwise or counter-clockwise. The motor shaft is rigidly connected to one end of the torsion bar, which is equipped at the other end with two positive unidirectional clutches. One clutch prevents clockwise, and the other counter-clockwise, rotation. If the motor is rotating in a clockwise direction and the proper clutch is engaged, the adjacent end of the torsion bar is stopped, thereby bringing the rotor to a stop in approximately 20 degrees. The potential energy stored in the torsion bar is then returned to the rotor in the form of a counter-clockwise impulse, and the motor attains virtually full speed in the new direction within three milliseconds.

Two methods are used to check the reversing time of the motor. For the simpler test, a black disk bearing a white index-line is mounted on one end of the motor shaft and photographed against a circular scale, using a high-speed motion-picture camera at approximately 4,000 frames per second. When the film is run through a projector at the normal rate of 16 frames per sec, the behavior of the disk is easily observed. When the motor is rapidly reversed, the backward speed is, for a short time, slightly lower than the speed before reversal because the ball-bearing assembly that supports the bar does not reverse its rotation as rapidly as does the motor shaft.

The second means of measuring reversal speed involves locking the far end of the shaft and applying direct current to one winding of the motor, while feeding alternating current to the other winding. Oscillations are thus induced in the rotor-torsion bar; and, by varying the frequency of the current, it is possible to determine the frequency of torsional resonance. The time required for the rotor to reverse can be easily computed, since it is one-half of one period of a torsional oscillation at resonance. The resonance in the system tested occurred at approximately 125 cycles per second, indicating (in a period of eight milliseconds) a reversal time of four milliseconds.

In the experimental model of the rapid-reversal mechanism built at the Bureau, the clutches are operated manually, but it is expected that, in normal use, the clutch mechanism will be operated by suitable electromagnetic controls. Although the studies thus far have centered around a small motor of about 1/75 hp, it is anticipated that motors of all sizes could be reversed rapidly by this technique, the speed of reversal being limited only by the mechanical strength of the various parts. Moreover, by use of suitable circuits, rapid reversal of motors can be obtained without producing large current surges in supply lines.

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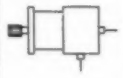
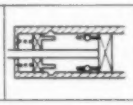
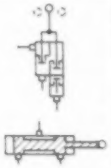
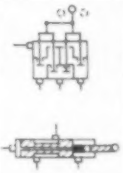
Standardized Hydraulic Symbols

Component	"A" Elementary Symbol	"B" Typical Full Symbol	Remarks
Pump — Power			"B" Symbols approximately to shape
Hand			
Tank			"B" Symbols approximately to shape, with filters, etc.
Cylinder — Simple			"B" Symbols should be shown in their their relative magnitudes, travels, etc.
Multi-travel			
Multi-volume			
With locks			Represent lock action diagrammatically if practicable

Component	"A" Elementary Symbol	"B" Typical Full Symbol	Remarks
Selector — 2-way			"B" Symbols should represent construction of the screw cock as shown
3-way			"B" Symbols should represent construction (Poppet and slide types of particular construction shown)
4-way			"B" Symbols should represent construction (Poppet and slide types of particular construction shown)
Complex	Both "A" and "B" symbols may have to be represented by diagrammatic representation of actual unit. In certain cases the "A" symbol can best be made up of two or more simple symbols with a note relating to their interconnection		

Pipe Line Conventions

1. One-way flow
2. Two-way flow
3. Emergency flow
4. Pipe connection-internal
5. Pipe connection-external
6. Pipes crossing

Standardization of Hydraulic Symbols

From "Standardization of Hydraulic Symbols" by H. G. Conway, The Engineer (London); December 30, 1949, page 772.

EFFORTS HAVE BEEN MADE by the British Standards Institution, in the publication "Graphical Symbols for General Engineering", to standardize methods applicable to at least the simplest and commonest elements of hydraulic or steam systems. Publication of Part I, "Pipes and Valves", of B.S. 1553 has already been published. These symbols may be suitable for the elementary type of circuit diagram but the complete or comprehensive diagram of an extensive hydraulic system requires more descriptive and less symbolic "symbols".

Three types of diagrams are needed for full comprehension of a complex hydraulic system: (a) A simple condensed diagram illustrating the principle of the system; (b) A complete detailed theoretical diagram showing the whole system in its entirety; and (c) A perspective drawing showing the physical location of all components in the machine, vehicle or aeroplane. It is the first two of these diagrams that concern the hydraulic engineer, and in which standardization is needed.

The elementary diagram should include only such items as are necessary to understand how the system operates. For example, whether a constant displacement pump with unloading valve or a variable delivery pump is used; whether the system is double acting or single acting; whether it is open or closed circuit.

It is therefore desirable to avoid refinements in the system, such as filters or restriction valves; or the duplication of items such as actuating cylinders, the system being equally comprehensible if one unit is shown when in reality, several are installed in parallel.

It is also unnecessary for the comprehension of the system's operation to have constructional features of the components revealed. A selector can be shown in its simplest rotary or slide form irrespective of whether it is fitted with poppet valves or not. An unloading valve can best be left as a block diagram since its construction does not normally affect the basic operation of the system. A portion of the recommendations for the main symbols of the elementary type and detailed symbols are shown above.

The complete diagram is equivalent to an electrical or radio wiring diagram and must be as complete and accurate as possible. Certain principles have been formulated for the preparation of this diagram and its symbols:—

(a) Every component in the system must be included.

(b) Components should be disposed on the drawing with regard to the arrangement on the machine or vehicle—that is, by arranging items to the right and left, top and bottom of the drawing, more or less as they are in the machine, yet within the inherent limitations of the drawing shape and size.

(c) Some regard should be paid

to the relative size of similar components—large and small actuating cylinders should be shown appropriately.

(d) Symbols should be chosen which illustrate the principal constructional features of the component. A poppet selector should be shown with valves, a slide selector with a slide. This should not, however, be allowed to interfere with the simplicity of the symbol that is necessary for easy understanding.

(e) While pipe-to-pipe line connections should not normally be shown all tee and cross, connections should be shown as faithfully as possible. Internal connections should be made with a dot, and the various symbols that represent a complete unit should be boxed with dotted lines.

(f) Pressure settings of relief valves, reducing valves and the like should be indicated close to the symbols.

(g) Pipe sizes should be indicated where possible.

Certain conventions that have been adopted for piping and to show the direction of flow are illustrated in the chart. Although B.S. 1553 recommends breaking one pipe where it crosses another, "looping" may be preferred to avoid mistakes, especially on small scale diagrams. In the case of printed diagrams, additional use may be made of multi-color printing to identify different circuits.

Effect of Fuel Immersion On Laminated Plastics

Abstracted from National Bureau of Standards, U. S. Department of Commerce, Washington 25, D. C. Technical Report 1436.

CHANGES IN WEIGHT and in the dimensional and flexural properties of laminated plastics immersed in three hydrocarbon liquids have been measured at the National Bureau of Standards as a means of evaluating the relative resistance of the laminates to different fuels. Constituting one phase of a continuing program on the properties of laminates for a wide variety of aircraft applications, the tests were designed to provide data needed in the preparation of specifications for those plastics found suitable for use as auxiliary (droppable) fuel tanks, radomes, instrument panels, and containers. The study involved the determination of the effects on representative laminated plastics of cyclic and continuous immersion in three fuels. The laminated plastics tested were: lignin and paper; melamine and cotton fabric; phenolic and asbestos fabric; phenolic and cotton fabric; phenolic and rayon fabric; phenolic and paper; polyester and cotton fabric; and polyester and glass fabric.

The cyclic immersion test consisted of 10 alternate 24-hour periods of fuel-immersion and air-drying. Weight, dimensional measurements, and flexural determinations were made after reconditioning at 77 F and 50 percent relative humidity for 48 hours. For the continuous immersion test, duplicate sets of specimens were immersed for 7, 30, 90, 180, and 360 days. On one set of specimens weight, dimensional, and flexural measurements were made immediately after removal of the specimens from the fuel and on a second set, after reconditioning for 7 days at 77 F and 50 percent relative humidity. The fuels used for these tests were heptane (an aliphatic hydrocarbon), toluene (an aromatic hydrocarbon), and SR-6, a representative aircraft test fuel composed of blended aliphatic and aromatic hydrocarbons.

Dimensional and flexural strength changes are the principal properties to be considered in many of the aircraft applications. Most of the materials may be used for this purpose. Considering only the 360-day continuous immersion test, a cotton-fabric phenolic exhibited the least changes in dimensions, and a paper phenolic laminate showed the smallest changes in flexural strength in the three fuels.

In all three fuels the weight changes of the majority of the laminates were less than 1.0 percent in the cyclic test, and did not exceed 1.5 percent after 360 days of continuous immersion. The largest weight changes were usually obtained in toluene. With very few exceptions, the length and width changes after either the cyclic or the 360-day immersion in the fuels did not exceed 0.1 percent.

The best weight and dimensional stability in all three fuels in the cyclic test was observed with the glass fabric unsaturated-polyester laminate. After the 360-day immersion test, the weight and dimensional changes were, in the majority of cases, equal to or higher for the samples that were tested immediately than for those that were reconditioned before testing.

The changes in flexural strength for the cyclic test and after the 360-day immersion test were, in most cases, negative. However, the losses were not over 10 percent for all of the samples.

The changes in flexural modulus of elasticity were, in the majority of cases, negative in the cyclic and the continuous immersion tests. In the cyclic test losses greater than 10 percent were shown only by the cotton-fabric unsaturated-polyester sample. After 360 days of immersion no losses greater than 10 percent occurred.

New Gas Engine Design

Abstracted from "New Type Gas Engine Installed on Gas Transmission Line," Power, April 1950.

RECENT INSTALLATION of a 1,600 hp gas engine and compressor set points up the apparent trend to re-evaluate and redevelop the spark ignition, gas engine. Compared to other fuels, natural gas is now in a favorable position in many parts of the country. More use is being made of by-product gases, and this type of engine should find more applications with the increase of natural gas transmission lines.

The important features of this gas engine are: spark ignition, two-cycle operation with uniflow scavenging, cam-operated exhaust valves, and higher compression pressures. This 8-cylinder engine has a 16 in. dia piston and a 16 in. stroke. It operates at 320 rpm and at a 77 psi bmep. There are two scavenging cylinders which gives a certain amount of supercharging. This scavenging pump increases the blowdown and charges the cylinder with a greater weight of air. As a result, a higher mean pressure can be developed without higher cylinder temperatures. The maximum pressure is about 280 psi, which is almost twice that of conventional gas engines.

There are a number of design details which are interesting. The cylinder liners are removable and have water jackets that are cast integrally into them. This construction reduces the possibility of cooling water leakage. The wrist pins are bolted to an inner wall inside the piston. This construction gives the pistons unbroken outer walls. These pistons are cooled by means of an oil spray which covers the upper inside part of the piston. In each cylinder there are two exhaust valves which are driven separately by rocker arms and a cam shaft. This valve gear is pressure lubricated. The connecting rods and crankshaft are forged. The ignition system consists of a low-tension magneto which distributes current to a high-tension coil located on each cylinder head. The fuel gas is supplied to the engine at 40 to 60 lb per sq in., depending upon its quality. There

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are three separate protective devices to guard against: (1) overspeed, (2) lubrication oil failure, and (3) excessive jacket water temperature.

This engine has been installed at the West Monroe Louisiana Station of the Tennessee Gas Transmission Company. Experimental testing of a three-cylinder experimental unit demonstrated fast starting, even firing on all cylinders at no load, and smooth running at low speeds down to ⅓ of the rated speed. Overload capacity beyond normal field requirements was observed and low exhaust temperatures measured at all loads. Good economy of this engine was reported.

Fuel Gage Indicates Weight Rather Than Volume

Abstracted from "Airplane Fuel Gage," *Electronics*, April 1950, published by McGraw-Hill Publishing Co., 330 W. 42 St., New York 18, N. Y.

LABORATORY ACCURACY is provided in a rugged electronic instrument that measures the weight of fuel, rather than volume. Designed for aircraft use, it employs the self-balancing bridge principle.

During the latter part of 1942, the disadvantages of the conventional float types of gasoline gage for aircraft became apparent. This gage employed a cork or hollow metal float to measure the level of the fuel; the information was then transmitted via gears, magnets and/or selsyns to an indicating instrument on the pilot's or flight engineer's instrument panel. Serious errors resulted from gasoline slosh, the volumetric expansion of gasoline with an increase in temperature, changes in supply voltage and wear in mechanical linkages.

In the effort to obtain a better fuel quantity gage for both military and commercial airplanes, this capacitor-type gage was developed. It makes use of the dielectric constant of gasoline, which varies of 1.85 to 2.3, depending on the temperature and the constituents of the gasoline.

The basic detecting circuit is a bridge with two capacitance legs and two inductance legs. One of the capacitance legs is a concentric tube capacitator that is mounted vertically in a cell of the gasoline tank.

Any change in fuel level, unbalances the bridge. The unbalance voltage is amplified and fed to the control circuit of a motorized potentimeter. Movement of the potentiometer arm again balances the bridge. Since its position is directly related to the amount of fuel, the potentiometer arm can be used to drive a dial pointer for direct indication of fuel level.

Variations in volume caused by temperature changes do not unbalance the bridge because such variations are almost entirely compensated for by specific gravity changes. Thus, the gage reading is truely proportional to weight.

Impact Stresses In Systems of Bars

Foreign abstract from "Impact Stresses in Systems of Bars", two parts, by Boris N. Cole, The Engineer (British), March 3, 1950 and March 10, 1950.

IN EXISTING discussions of impact waves engineers have adopted either the medium of operational calculus, or that of graphics. A non-calculus analysis is offered here to compute, within limitations, the maximum stress resulting in various systems of bars from a single impulsive blow. Since the main concern is to verify that the material will not be stressed beyond its elastic limit, only elastic waves are considered. Plastic deformations are excluded.

The fundamental equations for wave motion are taken as:

$$\text{propagation velocity} = a = \sqrt{\frac{E}{P}} \text{ ft/sec}$$

$$\text{particle, or surge velocity} = w = \frac{f}{a\rho} \text{ ft/sec}$$

where E is Young's modulus, f is the axial stress associated with a wave, and p is the density of material. With a compressive wave (f positive) the velocities w and a are co-directional. In a tensile wave, w and a are opposed.

In regions of wave superposition, it is assumed that component stresses and particle velocities may be added algebraically to give absolute values. The common surface between two bars is taken to have one velocity and equilibrium of forces is postulated at the same point.

With this in mind, it can be shown that at a change of section:

$$f_r = \left(\frac{k\,s - 1}{k\,s + 1}\right) f_i$$

and

$$_t = \left(\frac{2\,s}{k\,s + 1}\right) f_i$$

where f_t and f_r designate respectively the transmitted and reflected waves resulting from the incident wave f_i, k is the ratio of the areas, and

$$s = a_1\,p_1/a_2/p_2$$

The axial impact of two bars can now be investigated. Bar 1 is permitted to impinge with velocity V upon rigidly supported bar 2. The primary stresses are found as follows:

$$_1 \quad f = \frac{a_1\,p_1\,V}{K s + 1}$$

and

$$f_2 = k\,f_1 = \frac{k\,s\,a_2\,\rho_2\,V}{k\,s + 1}$$

The effects of wave reflections from the fixed end of the rigidly supported bar are sumed for n cycles in bar 2:

$$f = 2\left(\frac{1 - R^n}{1 - R}\right) f_2$$

using $R = (Ks - 1)/(Ks + 1)$. This stress will reach its ultimate value immediately before interruption by the return of f_1 after reflection from the free end of bar 1. For this condition:

$$n = > \frac{L_1}{L_2}\,\frac{a_2}{a_1} + 1$$

In Bar 1, the primary stress f_1 is increased by the addition of that portion of f_2 that is transmitted across the common surface each time f_2 is reflected in Bar 2. This process goes on until the return of the reflection of f_1 from the end of Bar 1 to cause interference. Hence in Bar 1:

$$f = \left[1 + k + \left(\frac{1 - R^m}{1 - R}\right)\right] f_1$$

$$m = > \frac{L_1}{L_2}\,\frac{a_2}{a_1}$$

From a consideration of joint conditions, it can be proved that any discussion of impact stress in space frames from the viewpoint of direct stress must lead to absurd results. However, the concept of shear waves, acting as compliments to waves of direct stress, brings the problem to a rational solution. Shear waves are handled in roughly the same manner of treatment as the direct waves.

The new wave method is an improvement in accuracy over the strain energy concept of impact and proves itself simplier than the methods of operational calculus. Conventional strain theory posulates that the whole of the impact system is motionless for an instant. Both shear and direct stress waves are disregarded. Curves can be drawn of the assessments of the strain energy theory analysis and the shock wave analysis of a given problem. The results of the two will have common asymptotes but variations will otherwise be considerable. The strain energy theory sometimes gives high values but more often, values lower than the true stress.

GENERAL PLANT VIEW of the new USAF Manufacturing Methods Pilot Plant in Adrian, Michigan. The plant has 17 acres under roof. During the last war Bohn Aluminum and Brass Corp., who occupied the plant, made aluminum extrusions.

Air Force Pilot Plant Open to Industry

SET UP UNDER THE SPONSORSHIP OF THE AIR MATERIAL COMMAND OF THE USFA, a new Manufacturing Methods Pilot Plant at Adrian, Mich., will serve as a testing and development proving ground for any American industry interested in the extrusion or forging of metals.

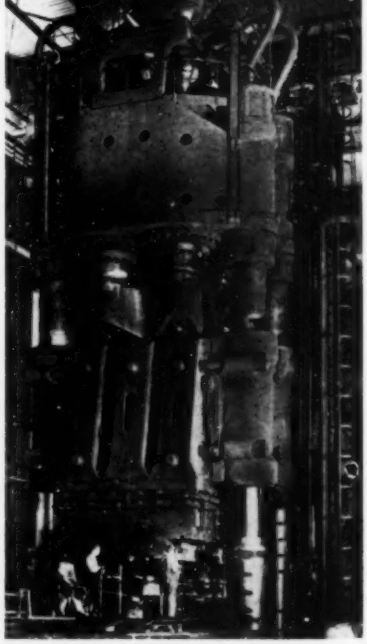
The Air Force, working with the Gerity-Michigan Corp., and Hydro Press, Inc., has gathered together the biggest and best in the way of equipment. Fifteen extrusion presses now installed range in size from 1,650 to 5,500 tons. Four forging presses being installed range from 600 to 15,000 tons, and a 75,000 ton giant is being built. These and other equipment and service, are available to industry on a "use time" fee basis. The big advantage to industry, of course, is that of having top facilities for research and development available, at no capital investment costs. Fees for facility use will be nominal ones designed to cover maintenance costs.

The new development plant should provide American industry with: (1) new extrusion methods for all types of metal, (2) increased knowledge of and use of the forging methods, (3) new design concepts, and new uses for extruded and forged parts, and (4) a broader basic knowledge of the effects of heat, pressure, and die design. The aircraft industry will probably be the most extensive user of the new pilot plant, but will by no means have a monopoly on the facilities. In an emergency, of course, the Air Force can take over facilities and channel development work along priority lines. During the last war the Air Force found U. S. industry short on forging techniques and equipment. From all indications the Russians are ahead of us now—thanks mainly to captured German equipment and technical talent. The Adrian project intends to change this situation.

This is the way the plan works. Company X is interested in metal forming and has a new idea to work on. It then contacts the Gerity-Michigan Corp., who runs the model plant for the A.F. on a contract basis. Air Force engineers confer with Co. X and G-M personnel to see whether the idea is feasible and can be scheduled. When the idea has approval Co. X moves in its own technicians to collect data as the idea is tested. Air Force technicians also keeps notes on progress and circulate the information gained freely among American industries, except in the case of secret projects.

The next big project at the new test plant will be the construction of larger presses. One 75,000 ton press that has been under study will be the largest yet built. It could forge a complete wing stabilizer with one die. This is important to the Air Force, which does not consider present forging, largely drop hammer work, satisfactory.

It should be emphasized that this



MADE IN GERMANY, this 30,000 ton press is now in Russia, along with the trained German technicians who have the basic knowledge to use it effectively. Presses such as these gave the Germans a big advantage in the last war. In making aircraft propeller blades, for example, the Germans were able to beat our top production by several hundred percent. The new pilot plant at Adrian will put us in a more competitive position.

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new development plant is not restricted to military projects. All types of manufacturing, within the limitations of forging and extruding, can use the facilities. Projects are not limited to light metals, but include all methods and materials where results may be obtained by use of greater pressures and heats.

Hydropress, Inc. of New York City, is installing the large German presses. The 15,000 ton cavity press will weigh about 1,000 tons when assembled. It requires a 30 ft excavation for the foundation. The four tie rods are 3 ft in diameter, 50 ft long, and weigh 65 tons each. The big press should be ready to use by January, 1951. Other equipment is ready for use now.

Rocket-Driven Stake "Anchors" Light Planes To Shorter Landing

ENGINEERS of Air Material Command and All-American Aviation are experimenting with an "anchor" that shoots itself into the ground, holding back a light plane or glider so that shorter runways can be used.

The "anchor" is a 28-inch tube, $1\frac{3}{8}$ inches in diameter. It is inserted into a holder near the tail assembly of the plane. Just before touching the ground during landing the pilot pushes a button that discharges rocket propellant in the steel tube driving it 18 inches into the ground at a 45 degree angle.

Attached to the anchor is a 200-foot stainless steel ribbon wound on a hydraulic brake that absorbs the energy at the time of the braking action and permits a smooth, easy stop.

An L-13 liaison plane, coming in at 75 miles an hour can be landed with the explosive anchor in 168 feet, rather than the usual 700 feet. The Air Material Command reports that in ground tests, the anchor withstood a pull of 25,000 pounds. There is little recoil shock when the rocket is fired. The steel ribbon can be used at least 100 times before replacement.

Proposed Exchange Program May Benefit Smaller Manufacturers

SMALL FABRICATING and other metal-working firms, and small businessmen generally will undoubtedly benefit from the State Department's proposed international science and technology exchange program, says John C. Green, director of the Commerce Department's Office of Technical Services.

The OTS now disseminates technical information originating in Government laboratories or resulting from government-sponsored work in private research centers.

The proposed science exchange program is set down in a 170-page report made by Dr. Lloyd V. Berkner, since 1933 a scientist on the staff of the Carnegie Institution of Washington. Berkner was hired last October by Secretary of State Dean Acheson to "study and report on the responsibilities and functions of the Department of State in connection with the national science policy of the United States."

The State Department intends to carry out all of Berkner's recommendations which include the setting up of a Science Office in the State Department and the employment of "high-grade" scientists as science attaches at the various U. S. embassies overseas.

Berkner claims that we can get more out of foreign countries than they can from us. Reason is that we excell in applied research and development, but that the Europeans are somewhat ahead of us in basic research.

France Uses the Free Piston Engine

PARIS—Free piston engines developed from Pescara patents are finding increasingly widespread employment in France.

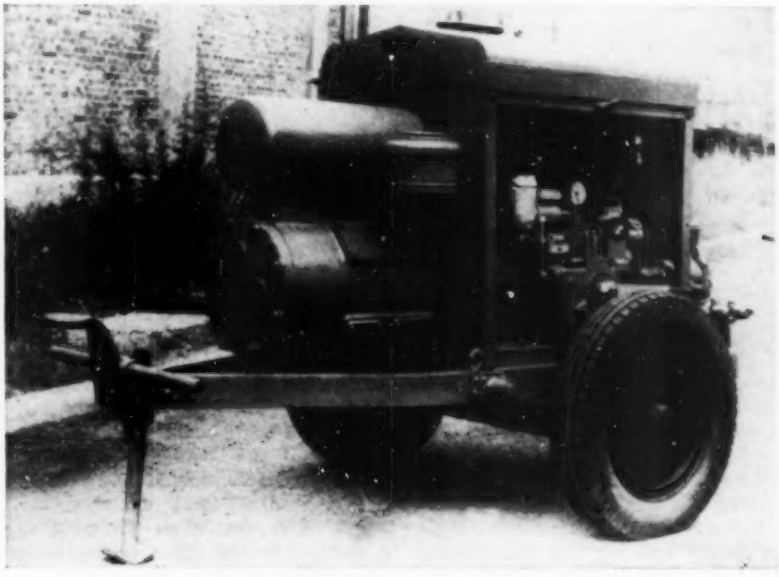
The Societe Industrielle Generale de Mecanique Applique, of Lyon, which now manufactures these engines, had sold over 1,000 free piston air compressors alone at the last tally. About 650 of these were being used on public works projects in France and its overseas territories. Another 100 or so were bought by the French Navy and the Electricite de France. The rest were scattered here and there in Great Britain, Belgium, Holland, Argentina, Portugal, Spain, Switzerland, and Central Europe.

A Pescara free piston engine consists of a two-stroke Diesel cylinder within which move opposed pistons. On either end of this cylinder are coaxial air cylinders fitted with pistons which form two continuous rigid assemblies with the engine pistons.

A lever mechanism synchronizes these two piston assemblies. They are driven outward by the explosion of fuel in the engine cylinder compressing the air in the air cylinders which then returns the pistons to the firing position.

Thus, piston rods, crankshaft, and flywheel are eliminated in the free piston engine, greatly increasing efficiency. Both friction and total weight are sharply reduced. Freed of the weight of conventional transmission mechanisms, the light pistons attain very high acceleration at the moment of combustion. Expansion of gases, therefore, is very rapid; heat losses correspondingly slight. The low inertia of the moving parts makes for quick starting in cold weather and at high altitudes. Since full consumption is in direct proportion to air output, the engines also save fuel.

The two most important applications of the free piston engine to date are as air compressors and as generators of electric power in connection with gas turbines.



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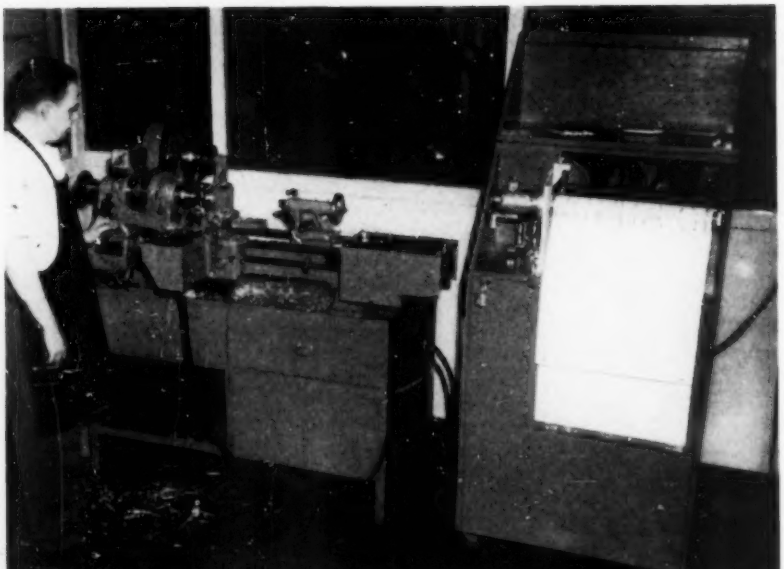


AN "ARMA-MATIC" ANALOGUE COMPUTER makes this engine lathe completely automatic. Holes punched in the wide paper roll control specific operations on the lathe. There is no theoretical limit to the number of operations possible.

A Forerunner of the Automatic Factory

IN A RECENT DEMONSTRATION by the Arma Corp., of Brooklyn, N. Y., visitors saw an engine lathe completely controlled by an analogue computer. A short stepped shaft was machined to an accuracy of ±0.0003 in. on the diameters and to within ±0.0025 in. on longitudinal dimensions by the completely automatic process.

The control unit, called "Arma-Matic", is a cabinet-housed assembly of induction potentiometers, step motors, electrical resolvers, synchro units, and other electrical components arranged to form an electrical analogue computing mechanisms.

In operation, a wide paper roll is punched, the holes conforming to conductor positions similar to the punching of cards on IBM machines. Thus, one line of holes across the paper roll will control a specific operation of the lathe. Upon the completion of that operation the paper roll is automatically turned to the next roll of holes in much the same manner as spacing lines on a typewriter. The next operation then begins.

There is no definite limitation on the number of operations that might be automatically controlled by the Arma-Matic. This control can also be applied to any make, size, or form of lathe and could also be extended to be applied to single or multi-spindle automatic screw machines, the analogue computer replacing the conventional cams and mechanical stops.

One of the great advantages of the new controller is that the perforated roll can be made in the drawing room or engineering department immediately after detailed design of the part has been completed. The perforated roll can then be filed for later use. When the job goes into the shop it is merely necessary to get the roll out of the file and mount it in the control unit of the lathe selected to do the job.

It is significant that the same roll can be applied for the control of any lathe regardless of the pitch of the lead and feed screws. Thus it is not necessary to perforate the roll to suit any given machine because the roll could be used on the Arma-Matic controlling any lathe in the shop.

Applications are not confined to lathes. Practically any machine tool in the shop can be equipped for this type of automatic control. It is being applied successfully to the cutting of cams on milling machines.

Although the initial equipments are being applied to machine tools in the shops of the users it is expected that the Arma-Matic equipped machine tools will be offered as original equipment. These computer—equipped machines will bring us still nearer to the concept of the completely automatic factory. Long considered only a dream of the theorist, these automatic factories now seem more plausable as industry develops the necessary basic tools and controls.

Bureau of Standards Improves Facilities

THE NATIONAL BUREAU OF STANDARDS has recently installed a new 2-high 16 x 24 in. rolling mill for use in a number of its research projects such as melting and casting of metals in vacuum, investigations of magnesium and aluminum alloys, including use of light metals for housing construction, and work on properties of ferrous metals, including stainless steel.

The mill takes 4-inch ingots or plate and is equipped for both cold and hot rolling. It has a pair of internally steam-heated rolls, permitting the rolling of magnesium alloys in a temperature range of 400 to 600 F. The rolls can be raised or lowered electrically. A hydraulic system, which provides a support for the roll bearings, makes a changing of the rolls an easy matter.

The Bureau of Standards also reports that important changes have been made in its foundry which now has one 650 lb tilting furnace, one 300 lb tilting furnace, one 200 lb lift-coil furnace, and one 100 lb vacuum furnace. In a separate fire-proof, dust-proof room are two sets of capacitors, and two new 250 kw generator units. This allows for simultaneous operation of the two furnaces independently. Plans are also underway for an overhead monorail system for the Bureau's molding room and other improvements.

Plastic Tower Helps Test Radar Antennas

A 30-FOOT HIGH, 300-lb plastic tower, believed to be the largest ever built, is being used today by Airborne Instruments Laboratory, Mineola, N. Y., to test the performance of the latest-type, faired-in (flush with aircraft skin) radio and radar antennas scheduled for Uncle Sam's newest, fastest planes.

Model airplanes up to 8 feet in wingspread and 50 pounds in weight are revolved in simulated flight at the top of the plastic tower. An amplifier, smaller than a cigaret package, installed in the model plane, transmits signals from the "airborne" antennas under test to recording mechanisms manned by AIL men on the ground.

Laboratory scientists had previously developed a radio antenna wave measuring technique known as the 105 system, but with the use today of ultra-high frequencies (sometimes called short-wave bands) on military

aircraft and commercial airliners, new testing equipment became necessary. The tower was built of plastics to eliminate confusing interference in radio waves being transmitted by the model aircraft while rotating in simulated test flights.

Uncle Sam, Industrial Designer

INDUSTRIAL ENGINEERS who have been able to read a copy of the omnibus small business aid bill (S3625) now in Congress, look upon it with disfavor. If it becomes law it would put the Federal government directly in competition with private industrial research laboratories and consultants.

The bill was introduced by Senator Lucas, for himself, and for Senators Maybank, O'Mahoney and Sparkman.

Principal provisions of the bill call for federal loan insurance and other financial assistance to small business. But the bill also provides for (1) a clearing house for technical and managerial information and (2) developmental engineering and research.

The clearing house proposal would merely legislate Commerce Department's Office of Technical Services into "legal" existence. OTS was established during the war by Presidential order and continued by such an order since. This bureau is already doing everything contemplated in the clearing house provision of the bill.

It is the research section (Sec. 503) of the bill that engineers look upon with jaundiced eyes. This section would authorize the Secretary of Commerce to undertake, through the National Bureau of Standards, other federal laboratories, nonprofit research foundations and educational facilities engineering and technical research on industrial, and commercial programs.

The Department of Commerce does not now have authority to carry on developmental work except under special conditions. Its chief research unit, the Bureau of Standards, is always careful to point out that they cannot do developmental work, and are limited to testing and basic research.

For a brief time (1946 and 1947) the Commerce Department had authority to contract for developmental research services. Its Industrial Research Division sponsored several hundred thousand dollars' worth of "industry-wide" research in private research centers and at universities. Congress refused to appropriate money for the Division's work in 1948.

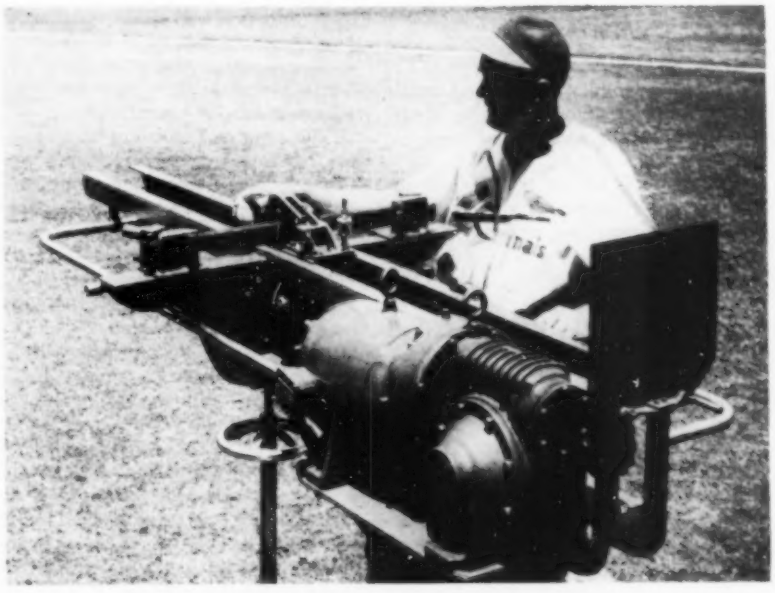
Processing Time For Color Film Cut

DAYTON—A new high speed technique which cuts, the time required for processing of color film from 90 to 20 minutes, and the printing time from 90 to 15 minutes, will soon be making color pictures as readily available as black-and-white. The kit of chemicals was developed under contract with the United States Air Force by the Ansco Division of the General Analine and Film Corporation of Binghamton, N. Y., to meet special USAF requirements. It has just been put through a series of rigid exhaustion and fading tests by the Engineering Division Photographic Laboratory at Air Material Command Headquarters, Dayton, Ohio, and so far the color quality resulting from the new process has shown itself to be equal to, or better than, that achieved through normal color photo-processing methods.

Photographic Laboratory engineers point out that the new process will considerably reduce the delay which now exists between the time the pictures are taken by photo airplanes and the time they are ready for inspection by photo interpreters. This time factor is of considerable importance from a military standpoint.

Basis of the new method is a prehardener which permits the processing to be done at 80 degrees instead of 68 to 70 degrees used in ordinary color processing. The hardener prevents the emulsion from becoming too soft at the higher temperature. All ordinary standard chemicals previously used in USAF color processing kits have been altered to meet new requirements. The new kit is the first ever developed which can be used for either paper or film, and requires the addition of only one special ingredient when used for film.

At present the supply of new kits is limited and the complete output of the manufacturer will be turned over to the Armed Services. The material will be available to commercial users as soon as military requirements have been met. Present kits are being made up only in the 3 and ½ gallon size. Service tests will be conducted by the USAF at Eglin and Lowry Air Force Bases, and if subsequent results prove satisfactory, the new method will be standardized.



No Sore Arm Here

THIS MECHANICAL BASEBALL PITCHING MACHINE can toss a ball every eight seconds. It works like a slingshot, with a gearmotor stretching a rubber band to furnish the power.

Things are now coming to the point where the great American sports lover can sit in his own living room and, with the aid of a television set, watch a machine play baseball. The man in the white suit works for a professional baseball team called the Cardinals—a corporation whose home office is located in St. Louis, Missouri.

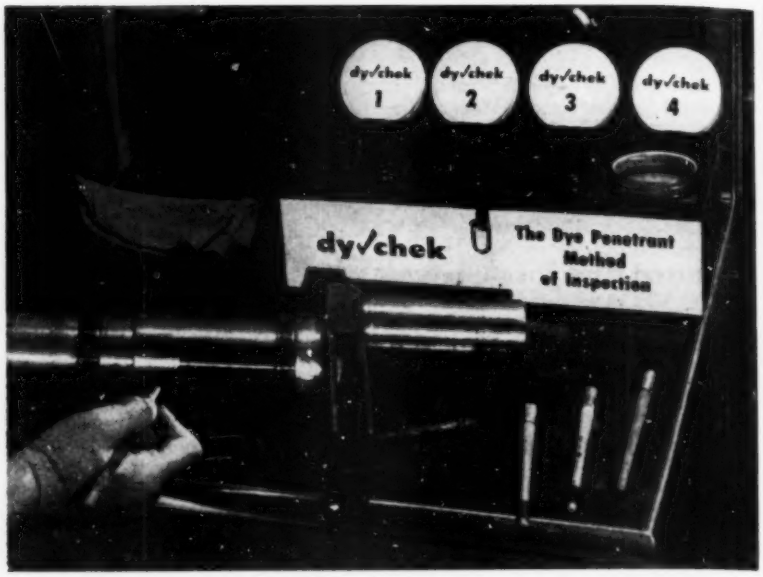
Northrop Develops New Metal Inspection Method

HAWTHORNE, CALIF.—Engineers at the Turbodyne Corporation, a subsidiary of Northrop Aircraft, Inc. have developed a new metal inspection process known as Dy-Chek, which promises great benefits to manufacturers and users of high-strength, precision-built machinery and equipment. The process, which requires only a paint brush, spray booth or dip tank, can test any metal within 20 minutes—sometimes faster, depending on the metal—for flaws. And without disassembling parts.

Actually, it does scientifically what older methods for quick inspection tried to accomplish—such methods as using dirty oil on hot forgings or the better known kerosene-and-white inspection. The need during turbine research at Northrop was for a quick, sure process to check for flaws while parts were still assembled. Even more important in checking high stressed parts, was need for an immediate foolproof check as to whether a scratch on a piece of metal was simply a scratch or a deep-rooted flaw. Before the dye penetrant method, suspected parts has to be disassembled and tested under any of the standard methods: X-ray, ultrasonic, magnetic or fluorescent penetration.

The process has several distinct advantages: It inspects and detects for the most microscopic, invisible surface cracks. It can give an indication which, after interpretation, reveals the extent, depth and width, and kind of flaw in the metal. It works on all metal, both non-magnetic or magnetic, with the possible exception of very porous grey iron castings. It gives no false indications. It requires no electricity, no bulky equipment, and a complete test kit is readily portable—needing only three bottles of special testing fluids.

The dye penetrant system employs three easy-to-carry chemicals; a penetrant liquid of unusually high capillary action and correspondingly low surface tension—colored a brilliant red. This is a super-saturated solution, which means that the dye is carried to every crevice of the metal wherever the base solution penetrates. There is also a volatile solvent which removes the penetrant and cleans the metal's surface; and a white developer which is a multipurpose, quick-drying solution. It provides a white background upon which the brilliant red flaw indicator shows up with photographic intensity, and at the same time draws out, "bleeds," the penetrant from whatever flaw it has penetrated.



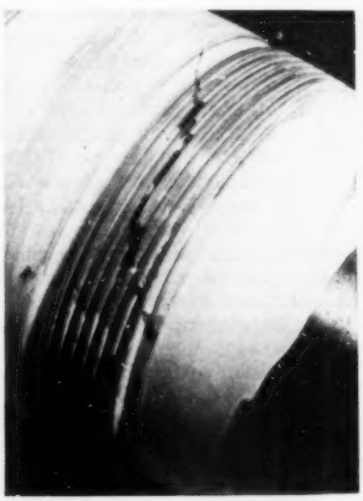
FOUR NUMBERED CONTAINERS in the small kit shown are used in the steps to be followed. Jar No. 1 contains surface cleaner and dye remover. No. 2 contains the red dye which gives the process its name. After the dye has been allowed to dry for a few minutes, it is removed through use of surface cleaner and dye remover from Jar. No. 3. Two containers of the identical surface cleaner and film remover are provided to "stationize" the inspection process. Jar. No. 4 contains the white developer. Once it is applied, scarlet lines will appear wherever cavities or cracks occur in the part.

The penetrant will find its way even into unseen, hair-thin flaws and work through these, from one abnormality to another, detecting surface flaws, crater cracks, lack of fusion and porosity. It inspects surfaces and sub-surface flaws so long as they have a surface manifestation, after parts have been welded, heat treated, ground, or after any machining process. It detects fatigue cracks in engines and in all types of machinery where preventive maintenance is necessary.

The penetrant has advantages over standard inspection methods in that it works equally well on magnetic and non-magnetic metals—on steel, stainless steel, nickel, nickel alloys, magnesium, titanium, copper, brass, aluminum, aluminum alloys and cobalt.

The red penetrant is applied to the metal surface and allowed to penetrate for from 5 to 15 minutes. For maximum sensitivity to very small flaws, the parts should be warmed up to 100 to 150 F. Usually, however, if the metal feels room temperature to the hand, it is warm enough for good results.

The penetration time depends upon the extent and thinness of the suspected flaw. A very tight flaw may require 15 minutes' penetration time-depending on the metal under test. The



HERE THE WHITE DEVELOPER has been applied to the crankshaft, and the dye penetrant has revealed a distinct fissure which could have resulted in a costly accident. The entire inspection operation took only a few minutes and was accomplished with a portable inspection kit. Dy-Chek is manufactured by the Dy-Chek Company, a division of Northrop Aircraft, Inc., 1515 East Broadway, Hawthorne, California.

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volatile solvent is then used to wash the area completely clean of the red penetrant. Once dry, the area is lightly painted—either spray, brush, or dip—with the white, fast-drying developer. As it drys white, giving a white background, this chemical "bleeds" or "draws" the red penetrant out of the metal—out of subsurface non-conformities. The outline of the flaw begins to show vividly against the white developer background almost immediately. Through practice and use, an inspector can estimate the extent—depth and width of the flaw by the shape of the red path which shows up on the white developer.

ASME Nominates New President

J. CALVIN BROWN, of Los Angeles, Calif., owner of the firm bearing his name in that city, has been nominated as the 1951 president of The American Society of Mechanical Engineers. Mr. Brown heads a slate of new ASME officers, including four regional vice presidents and two directors-at-large, submitted by the society's nominating committee. Since only one name is presented for each office, nomination is tantamount to election.

Formal election will take place in the Fall by letter ballot of the membership. The new officers are to begin their terms at the conclusion of the ASME annual meeting in New York next December. Mr. Brown will succeed James D. Cunningham, president of Republic Flow Meters Co., of Chicago.

A mechanical engineer and attorney at law, specializing in patent, trademark and copyright litigation before the United States courts, Mr. Brown became a member of the society in 1928. He served as manager of the ASME in 1943-44, and vice president, 1945-47 and 1947-49, and has been active in the affairs of the Southern California Section of the society.

Regional vice presidents nominated are: Henry Reginald Kessler, manager, Republic Flow Meters Co., New York; Stephen Dewey Moxley, vice president, American Cast Iron Pipe Co., Birmingham, Alabama; Dr. John T. Rettaliata, dean of engineering, Illinois Institute of Technology, Chicago; Carl J. Eckhardt (renomination), professor of mechanical engineering and superintendent of utilities, University of Texas, Austin.

Nominated as directors-at-large: Lionel J. Cucullu, assistant to chief engineer, New Orleans Public Service, Inc., New Orleans, La.; and Harold E. Martin, district manager, The Babcock & Wilcox Company, New York.

Unification of Nut And Bolt Standards In Final Stages

RECOMMENDATIONS FOR STANDARDIZATION of bolt and nut sizes in the U. S., Canada, and Great Britain will probably be accepted by leading societies of each country involved. The standards were agreed upon by delegates several months ago.

The recommendations, which also would involve some changes in United States practice, must be approved by technical associations corresponding to the American Standards Association, the American Society of Mechanical Engineers and the Society of Automotive Engineers in this country, to become effective in the co-operating countries. In turn, the ASME and SAE must approve them through appropriate committees and the ASA through its standards council.

The proposed standards for bolt and nut sizes that have emerged as a result of the conference, follow the accord already reached in the case of screw-thread standards and, if adopted, will make screws and nuts interchangeable on nearly all machinery, equipment, vehicles and other mechanical apparatus made in the inch-using countries. This is held probable, since it has been customary for all members of the British Commonwealth to adopt the standards approved by the top British technical group.

A GIANT INJECTION MOLDING MACHINE for plastic products presses out thermoplastic parts up to 200 ounces in weight. The big machine was made by the Watson-Stillman Co., Roselle, N. J.

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PRESIDENT TRUMAN has named Dr. Archie M. Palmer as chairman of the new Government Patents Board. Dr. Palmer is also Director of the Patent Policy Survey of the National Research Council and former president of the University of Chattanooga.

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A PROGRAM FOR PLASTICS EDUCATION has been issued by the SPI in the form of a 16 page booklet. The program is to provide a guide to colleges who want to set up specialized courses in plastic education. Copies of the booklet are being mailed to all technical schools.

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AUTOMOBILE HORNS can be made less raucous and yet just as loud, according to an Armour Research Foundation acoustics expert. His formula: cut out some of the high notes.

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GENERAL ELECTRIC'S RESEARCH LABORATORY will soon be moving intc complete new quarters. The new laboratory structure, situated near Schenectady, N. Y., totals 5 buildings with a usable area of 185,000 sq ft. Most of the area will be "universal space", with movable partitions that permit labs of any needed size.

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SEVENTY-FIVE SCIENTISTS from Southern Universities will spend this summer carrying out research in the Atomic Energy Commission laboratories at Oak Ridge, Tenn. For the most part they will be concerned with research on peace-time aspects of atomic energy.

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FRONT SURFACE MIRRORS are recommended for precise reflection by the Army Engineer Research and Laboratories at Fort Belvoir, Va. For 75 cents the OTS, Dept. of Commerce, Washington, D. C., will furnish a summary of studies on "Silicon Monoxide Front Surface Mirrors".

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A PORTABLE INSTRUMENT that measures the thermal conductivity of wet or dry solids within ten minutes, was reported at the semi-annual meeting of the ASHVE, at Muskaka, Ontario.

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Silicone News
DC
SILICONES
Dow Corning Silicone Insulation Saves Over $8,000 a Year on 4 Motors!
In the Saginaw Plant of Eaton Manufacturing Co., the output of four of these machines is stepped-up by stopping the 5 h. p., 3600 r.p.m. motors with a jolt of direct current as soon as 5 operations are performed on a small automotive part. With 105 starts and stops per hour, Class A insulated motors had an average life of only 8 days—32 failures for each of the 4 motors or a potential $10,000 a year in rewind costs alone.
Rewound with Dow Corning Silicone (Class H) Insulation, these motors now have a minimum life of 6 months; save Eaton a potential $8,000 a year plus a few hundred man hours of labor and some 40,000 parts that would have been lost due to down time.
DOWN TIME AND LOST PRODUCTION DUE TO MOTOR FAILURE ARE NO LONGER NECESSARY! Prove it in your own plant by having your hardest working motors rewound with Dow Corning Silicone Insulation. Call our nearest branch office for technical assistance or write to Dept. C-16.
Dow Corning Silicones Mean Business
DOW CORNING CORPORATION
MIDLAND, MICHIGAN
Atlanta • Chicago • Cleveland • Dallas
Los Angeles • New York
In Canada: Fiberglas Canada Ltd., Toronto
In Great Britain: Albright and Wilson, Ltd., London
FIRST in Silicones
Dow Corning

Among engineering materials the performance of *Dow Corning Silicones* sounds as fabulous as the tall tales told of Paul Bunyan or Mike Fink who claimed to be half-man, half-alligator . . .

The tall tales of the old frontier are reflected in skyscrapers . . . jet planes that disappear before you hear them . . . materials that are half inorganic, half organic. Fact has become more fabulous than folk tales. Here are a few of the fabulous facts about Dow Corning Silicones:

- DC Silicone (Class H) Insulation has at least 10 times the life of the best electrical insulating materials previously available. It also increases the power per pound ratio of electrical equipment by 100%.
- DC 44 Silicone Grease has at least 10 times the life of organic greases.
- Silastic*, the Dow Corning Silicone rubber, is serviceable at temperatures from −100 to +500° F.—an impossible assignment for any other kind of rubber.

The consumer is shopping for quality at a price. Management demands longer life, increased production and lower maintenance costs.

YOU NEED DOW CORNING SILICONES TO MAKE THE "IMPOSSIBLE" PRACTICAL

Write today for technical data on

DC SILICONE GREASES

Including relative bleed, evaporation and oxidation resistance of Silicone and organic greases, penetrometer, solidification and flash points; performance and application data.

DC SILICONE (CLASS H) INSULATION

Including results of accelerated life testing of Silicone (Class H) and Class B insulated motors; service life data on rewound equipment; and data on performance of equipment designed to capitalize on exceptional heat and moisture resistance.

*T. M. Reg. U. S. Pat. Off.

DOW CORNING CORPORATION, MIDLAND, MICHIGAN
Please send me ☐ Grease data sheet No. X-18.
☐ Silicone Insulation data sheet No. X-18A.

Name_____
Title_____
Company_____
City_____ Zone____ State_____

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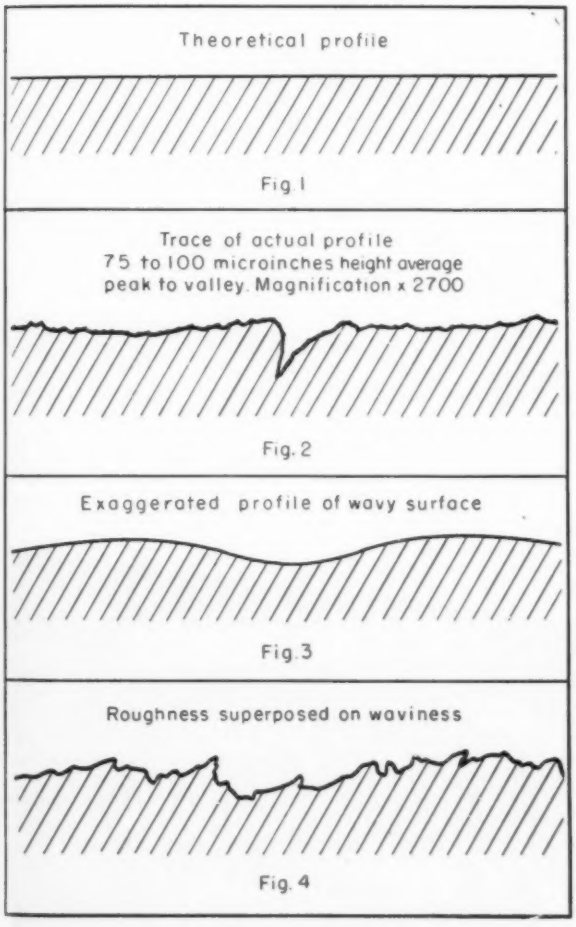
Surface Finish Definitions and Standards

Based on the Surface Finish section of the new SAE Automotive Drafting Standards, to be published by the Society of Automotive Engineers

ON ENGINEERING DRAWINGS, the "nominal" surface is represented by a line if shown in profile (Fig. 1), or by its boundary lines or contours if shown in a plan view. Practical surfaces do not possess the perfect smoothness represented by the drawing. Actually, a surface is a three dimensional boundary having not only a length, breadth and degree of curvature, but also height as shown in Fig. 2. The actual surface is composed of peaks and valleys of varying lengths, widths and heights with still finer peaks and valleys on the flanks of the major ones. Usually the engineer is concerned only with the major peaks and valleys. Following are definitions of surface finish terms.

PROFILE. The profile is the contour of any specified section through a surface, as shown in Fig. 1-4.



ROUGHNESS. Roughness is the relatively finely spaced surface irregularities, as shown in Fig. 2. On surfaces produced by machining and abrasive operations, the irregularities produced by the cutting action of tool edges and abrasive grains, and by the feed of the machine tool, are roughness. Roughness can be considered as being superposed on a wavy surface as shown in Fig. 4, but roughness in itself does not alter the "trueness" of the surface.

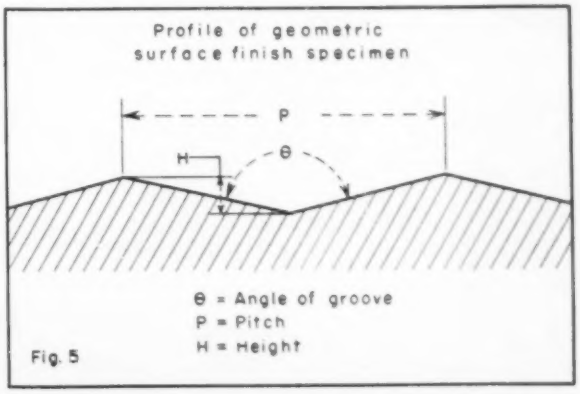
WAVINESS. Waviness is the surface irregularities which are spaced further apart than the roughness. On machined surfaces, such irregularities may result from machine or work deflections and vibration. Irregularities of similar geometry may occur from warping, strains, or other causes. (See Figs. 3 and 4.)

MICROINCH. A microinch is one millionth of an inch, 0.000001 in. It is the unit height of roughness. An abbreviated form, mu in., or symbol, μ in., may be used to signify microinches.

LAY. The lay is the direction of the predominant surface pattern. (See Figs. 13 to 14.)

FLAWS. Flaws are irregularities which occur at one place, or at relatively infrequent intervals in the surface—a scratch, ridge, hole, peak, crack or check.

ROUGHNESS STANDARDS. To correlate all surface finishes, a set of preferred standard roughness values has been developed. These values are based on geometric surface finish standard specimens having the profile shown in Fig. 5, and carefully constructed with known and constant spacing, angles, and roughness heights. These are expected to be adopted by the SAE as primary standards for surface finish control. The standard roughness height values are as follows:

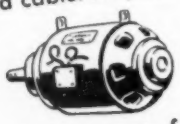
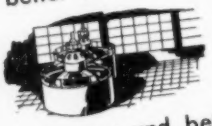


SAE Roughness Height Values—Microinches

1	5	13	32	80	200	500
2	6	16	40	100	250	600
3	8	20	50	130	320	800
4	10	25	60	160	400	1000

(*continued on page* 165)

The right wire for the right job...

Technological advances in many industries constantly require new developments and new applications of wire and cable. A space-saving wire design has made possible smaller, more compact  motors. A new and better form of heat resistance in turn prevents failure in generators.  Again, if a sharp turn can be given a wire, a new and better clock may be possible. New abrasion characteristics may mean milady's iron won't fail. A city's safety is made more certain because a solenoid sets new records in performance. As simple a thing as better shielding in a wiring harness can be the secret of clearer radio and television reception. Auto-Lite Engineers, drawing on their many years of automotive experience, can answer such problems. For more complete information on the "right wire for the right job" write on your company letterhead to

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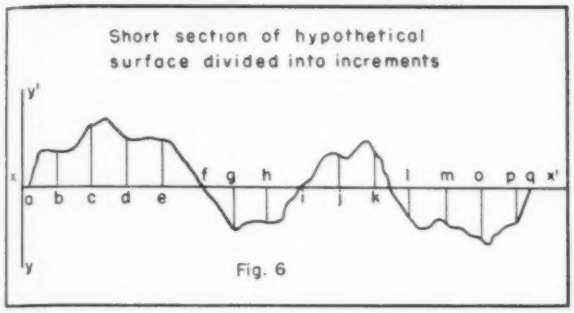

Fig. 6

The units in which these values are designated are average microinches—the arithmetical average of absolute deviation from the mean surface. Fig. 6 represents a short section of a random surface greatly magnified. Line X-X′ is the mean surface. The average value is:

$$H_{avg.} = \frac{y_a + y_b + y_c + y_d + y_e + y_f + y_g \ldots + y_n}{n}$$

where n is the number of increments, and the height measurements are made without regard to sign.

For the standard specimens themselves, the average value is one-fourth the height "H" in Fig. 5. The recommended inspection procedure is to use a suitable stylus-type surface roughness measuring instrument in conjunction with the standard specimens and the work. The dial figures should not be used directly because of possible variations in calibration. All measurements should be taken across the direction of the lay unless otherwise specified. For rapid checks, a calibrated pilot sample of the work carrying the proper surface may be used. If the limits are not too exacting, manual feel and visual comparison are satisfactory.

SURFACE FINISH SYMBOL. The approved symbol for designating surface finish is shown in Fig. 7. The point of the symbol can be placed on the line representing the surface, on the witness line, or on a leader line pointing to the surface. The angle between the two legs of the symbol is 60 deg, and the long leg is to the right and erect. The short leg, which may be increased in width, is approximately $\frac{1}{3}$ the length of the long leg. The top extension

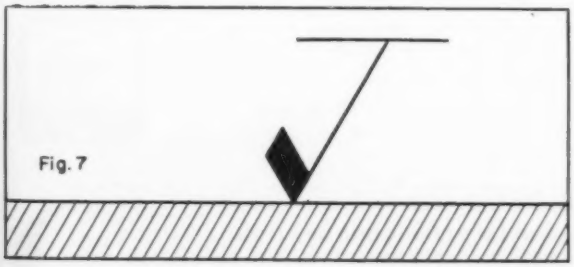
Fig. 7

line is approximately as long as the long leg, and extends about $\frac{3}{4}$ to the left and $\frac{1}{4}$ to the right of the intersection. The size of the symbol should be uniform on any drawing.

WAVINESS STANDARDS. The following preferred waviness height values have been adopted by SAE:

Waviness, inches

0.00002	0.00007	0.0003	0.001	0.005	0.015
0.00003	0.0001	0.0005	0.002	0.008	0.020
0.00005	0.0002	0.0008	0.003	0.010	0.025

The waviness width is not standardized but can be specified as shown later. Allowable limits of waviness values become particularly important in close tolerance fits and in cylinder bores. For waviness greater than 0.002 in., a surface gage is sufficiently accurate. For finer waviness, bluing gages can be used; however, the measurement and designation will be in percent of bluing rather than in inches of height. The commonly designated values are 50, 75 and 90 percent bluing. For extremely fine lapped or polished surfaces, optical flats can be used. The waviness designated for optical flats may be in inches or specified by a note, as shown in Fig. 8. The ordinary stylus-type instruments designed to measure roughness cannot be used to record waviness. However, instruments to measure waviness have been developed. Waviness can be controlled by using accurately aligned machines and care in processing.

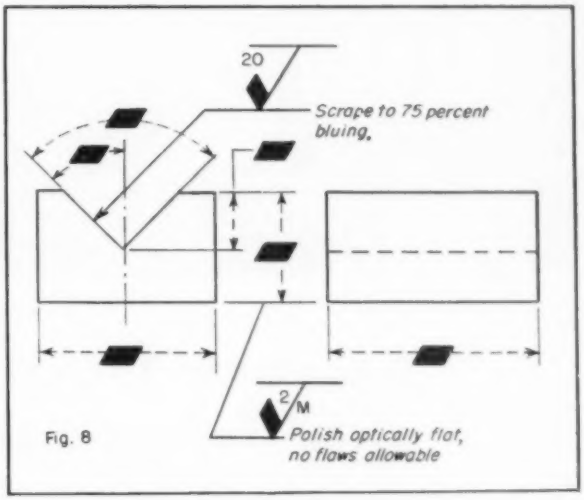

Fig. 8

DESIGNATION OF ROUGHNESS HEIGHT. The roughness height value is placed adjacent to and on the inside of the long leg of the surface finish symbol, as shown in Fig. 9. When upper and lower limits are desired, the upper limit will be placed above the lower. If a desired roughness exceeds the limit of the standardized values, the notation average should be appended to the roughness value.

DESIGNATION OF WAVINESS HEIGHT. The waviness height value is placed above the extension line (Fig. 10).

DESIGNATION OF ROUGHNESS AND WAVINESS WIDTH. The roughness width value is placed to the right of the lay symbol, as shown in Fig. 11. The waviness width value is placed to the right of the waviness height value preceded by a hyphen.

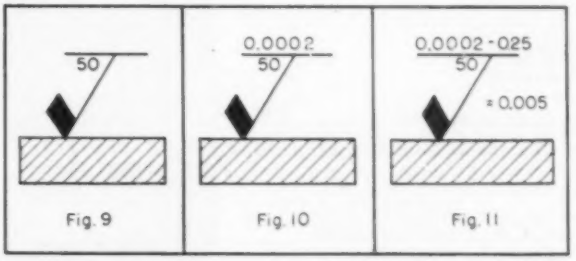

Fig. 9 Fig. 10 Fig. 11

(*continued on page 167*)

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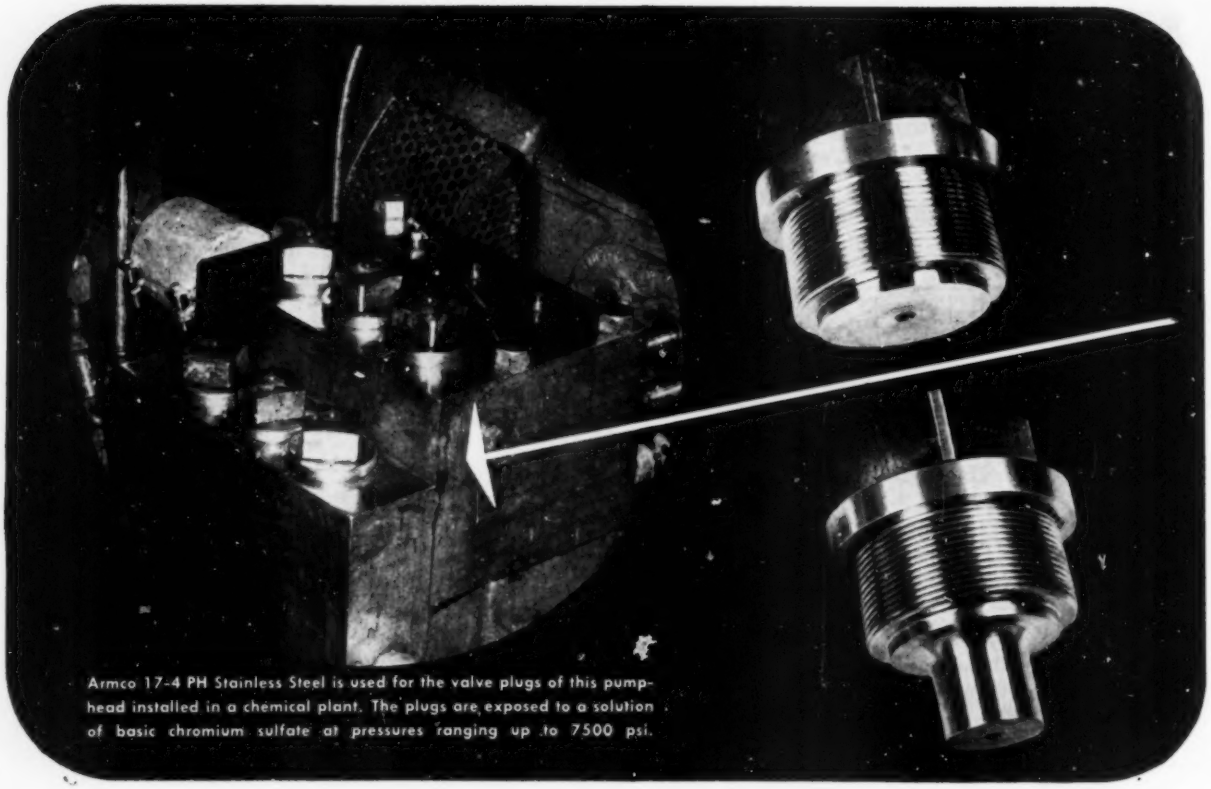

Armco 17-4 PH Stainless Steel is used for the valve plugs of this pump-head installed in a chemical plant. The plugs are exposed to a solution of basic chromium sulfate at pressures ranging up to 7500 psi.

Heat-treat this stainless for hardness after finish-machining

New Armco 17-4 PH Stainless Steel (a precipitation hardening grade) offers you many cost-saving advantages. One of the most important is the ***low-temperature*** heat treatment that gives your products hardnesses of Rockwell C40 to 45. Yield strengths range from 170,000 to 200,000 psi.

Heat treating in the low-temperature range of 850° to 900° F. does the job.

This means you can finish-machine precision parts ***before*** hardening. In fact, you get rid of finishing problems all along the line. There is no danger of cracking or distortion. No scaling either, and no stress-relief treatment is needed. Only an easily-removed heat tint results from the hardening operation.

Both the machining characteristics and corrosion resistance of Armco 17-4 PH Stainless Steel closely approach the 18-8 grade.

Production men report excellent results with Armco 17-4 PH in the production of gears, cams, shears, pinions, chains, valve plugs, shafting, special nuts, bolts, fasteners, and precision instrument parts.

If your products need high strength and hardness, or they can be made more salesworthy with the good corrosion resistance offered by Armco 17-4 PH, write for more information about this newest Armco special-purpose steel. Just fill in the handy coupon and drop it in the mail.

see our CATALOG in SWEET'S FILE for PRODUCT DESIGNERS

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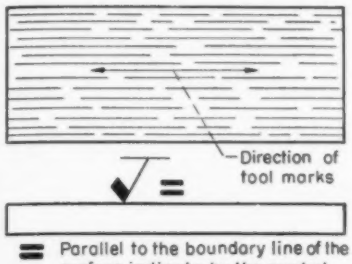
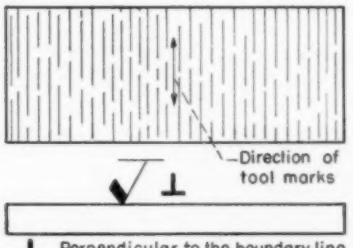
DESIGNATION OF LAY. Figs 12-17 are plan views of hypothetical specimens, the front elevation views of which would bear the illustrated standard lay symbol on the top line. The lay symbol is used only if considered essential to a given class of work. It is placed under the extension line of the symbol and to the right of the long leg.



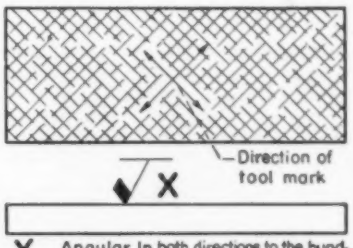
= Parallel to the boundary line of the surface indicate by the symbol Example: Parallel shaping, end view of turn or OD grind.

Fig.12



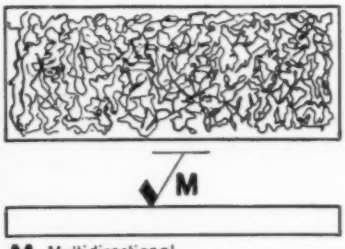
⊥ Perpendicular to the boundary line of the surface indicated by the symbol. Examle: End view of shaping, longitudinal view of turn and OD grind.

Fig 13



X Angular in both directions to the bundary line of the surface indicated by the symbol. Example: Side wheel grind, and traversed end mill.

Fig.14

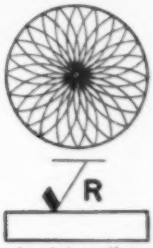


M Multidirectional. Example: Lap, superfinish.

Fig.15

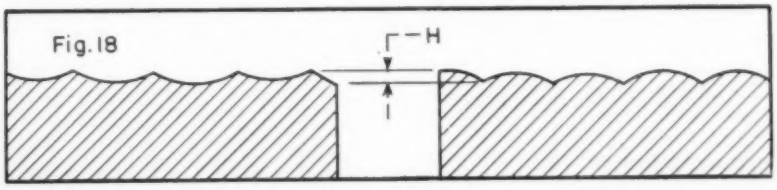
C Approximately circular relative to the center of the surface indicated by the symbol. Example: Facing.

Fig.16



R Approximately radial relative to the surface indicated by the symbol. Example. Surface ground on a turntable, fly cut and indexed on a mill.

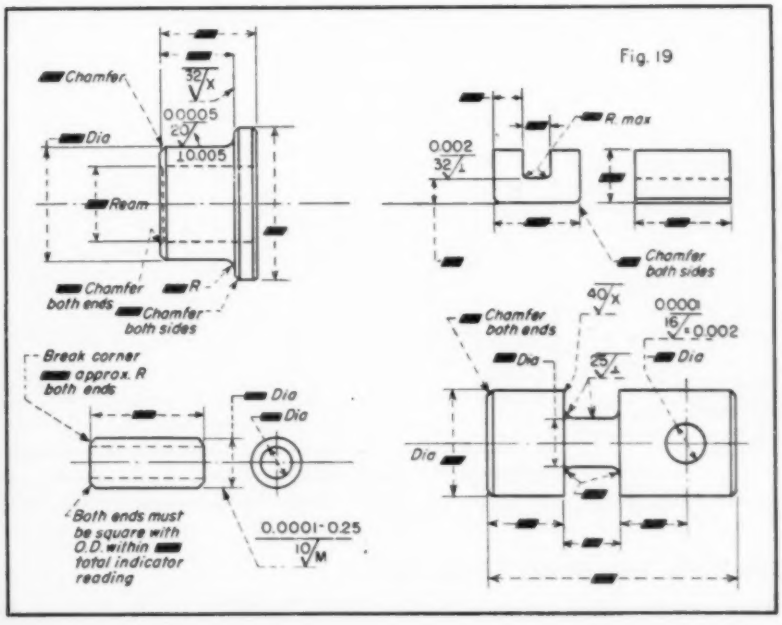
Fig.17

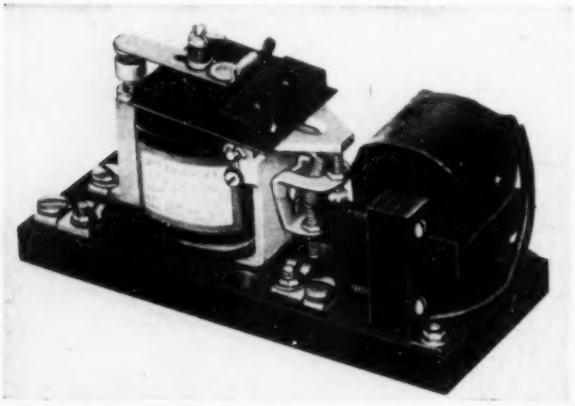


THE USE OF NOTES. Where the symbol is used on a plated or coated surface, a note should be added to signify whether or not the control applies to the base metal surface or to the plating or coating. If it is desired to control all surfaces, the following note can be used: "Surface finish on all machined surfaces shall not exceed XXX microinches unless otherwise specified."

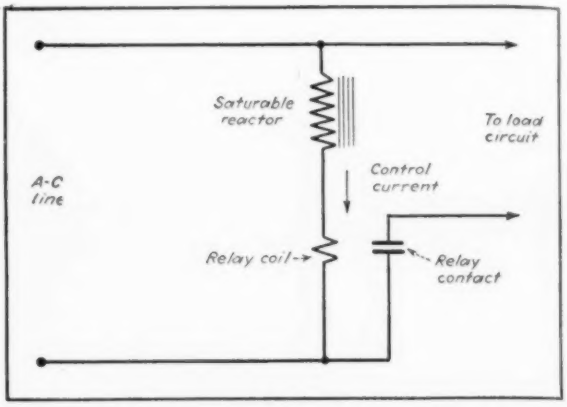
Depending on the function of a particular surface, minor flaws are generally permissible. When flaws are not allowed, a special note should be made, as shown in Fig. 8. In some applications, it is desirable to control carefully the character of a surface with respect to bearing area. As shown in Fig. 18, two surfaces with the same measured roughness can have entirely different characters. It is therefore occasionally necessary to indicate the processing procedure by a special note. Fig. 19 is a composite example of the use of surface finish designation.

EDITOR'S NOTE: These Surface Finish Standards should also be used in conjunction with the article "Surface Finish and the Designer" by Roy P. Trowbridge, to be presented in the September issue of PRODUCT ENGINEERING.





CHATTERPROOF RELAY ASSEMBLY consists of an a-c relay and a saturable reactor mounted together on an insulating base and provided with screw terminals.



IMPEDANCE IS MUCH GREATER in the saturable reactor than in the relay coil. Changes in the relay have little effect on the total circuit impedance and hence on circuit current.

Voltage Change Trips Relay Without Chatter

Relay opens when a-c voltage drops below minimum. Open and close voltages have small differential, giving accurate current control.

This relay was originally designed to protect single phase motors from line voltage drops and resultant current surges. Operation is positive, without chatter. The contacts open cleanly when the voltage drops below the present minimum.

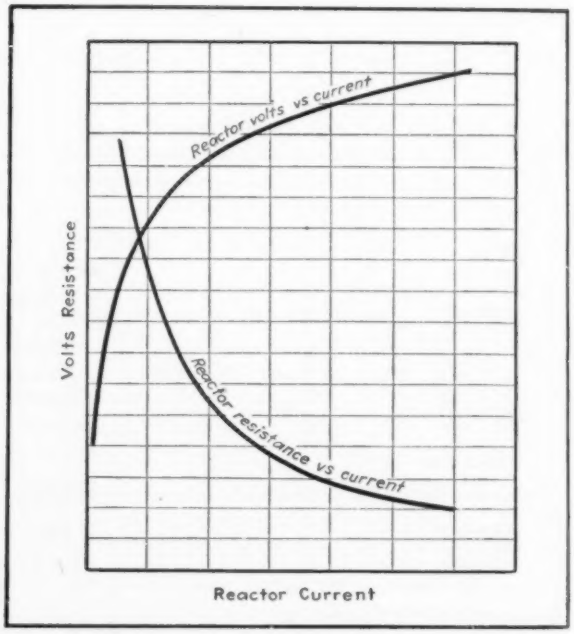
Close on-off voltage differential without chatter is difficult with ordinary relays. The impedance of the magnet coil changes when the relay opens or closes. When line voltage drops to the kickout setting, coil current no longer holds the contacts closed. The relay opens, making an air gap in the magnetic circuit and decreasing electrical impedance. Control current increases again pulling the relay closed again. This is chatter.

The new relay has a very large reactor in series with the relay coil. The changes in impedance caused by relay operation are very small compared to the impedance of the saturated reactor. Since control current is only minutely effected, chatter is eliminated.

Under operating conditions the large reactance in series with the relay is saturated. Small voltage changes result in wide control current variations. Fine control adjustments are possible. Standard relays with 115 v coils open at 95 and close at 105 v. Special 5 percent differentials are available. Maximum voltage is 20 percent above pick-up.

Specifications: S.P., N.O., 115, 230 and 440 v, 60 cycle. Contact ratings 25 amps, non inductive, 230 v. Voltage differential of 10 percent is standard.

Ward Leonard Electric Co., 115 MacQuestion Pkwy. South, Mount Vernon, N. Y.



SATURATION comes with increasing current through the reactor. In the relay operating range, a small voltage change gives a large control current change.

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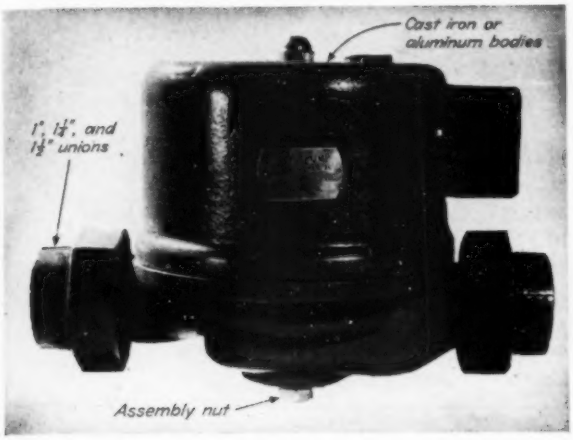
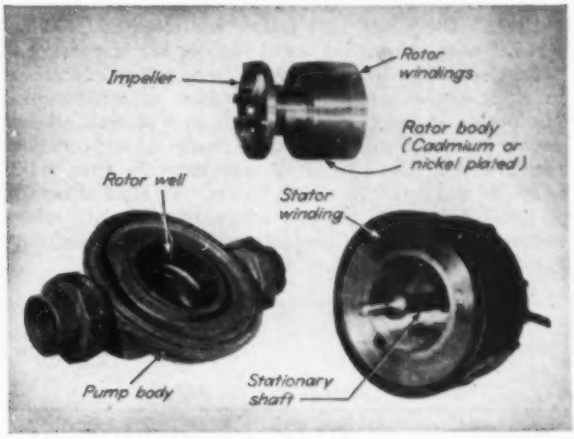
High Head Pump Has One Moving Part



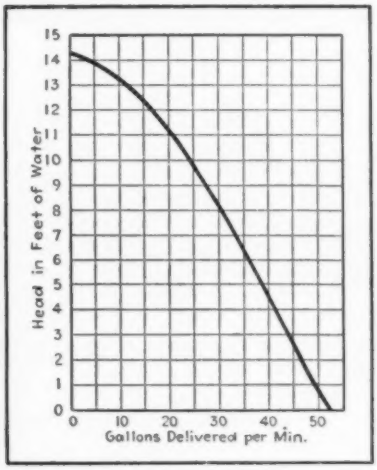
NO LUBRICATION is required in these centrifugal pumps delivering 10 gpm, 13 ft head, at 1/6 hp. Removing the bottom nut disassembles the 3 piece motor pump. Sizes up to 50 hp can be made. Windings are airtight; motor protection is automatic.

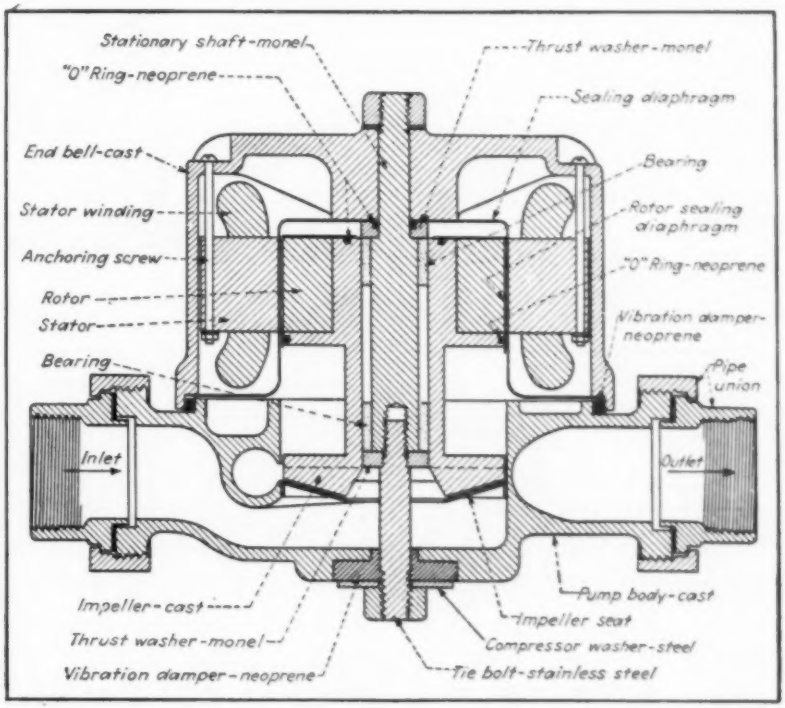


ONE MOVING PART includes pump, impellor and electrical rotor and rotates inside the stator on the stationary shaft. Stainless steel diaphrams hermetically seal rotor and stator. Almost any fluid can be handled with Graphitar or bronze bearings.

Bronze bearings are used on oil service in these pumps, while Graphitar bearings are standard for other liquids. The assembly weighs 30 lb and carries a two year guarantee. Cast iron assembly has union fittings, but the aluminum unit is flange connected.

Mono Products, Inc., 810 N. Hancock St., Philadelphia 23, Pa.





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CONTINUED ON PAGE 170

Cathode-Ray Indicator Graphs Electrical Impulses

This instrument provides a convenient means for measuring and continuously showing any phenomena which can be translated into electrical impulses. As a general purpose unit, it is designed to replace custom built models on as wide a range of applications as possible.

With different setups, the instrument becomes a pH meter, an electrocardiograph, a pressure indicator or a strain analyzer. The screen will follow any electrical impulses, whether static, fluctuating, or of high or low impedance.

Results are displayed on the screen of a 5 in. cathode-ray tube. It allows continuous observation of non-recurrent phenomena at sweep speeds of 0.01 to 5 seconds. A two decade potentiometer, calibrated with a self-contained standard cell, provides direct measurements from 0.001 to 15 pH and 0.1 millivolts to 5 volts. An input attenuator of 1,000 megohms p.ovides overlapping ranges of 1 micro-microampere to 10 milliamperes and 1 millivolt to 100 v per centimeter deflection.

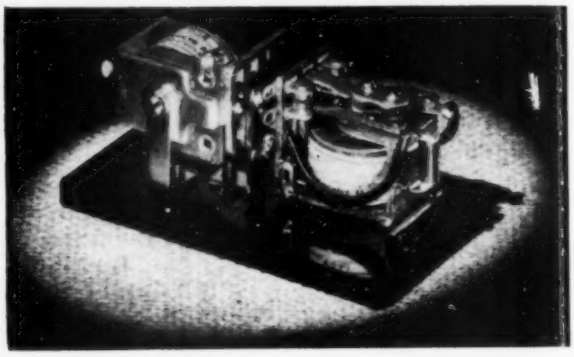
Analytical Measurements, Inc., 585 Main St., Chatham, N. J.

Circle No. 3 on Reader Service Card

Latch Holds Relay Armature Closed Without Current

A new mechanical latching relay utilizes midget relays as basic components. It is used where current is to be applied to the coil momentarily, the mechanical latch holding the armature closed until the latching coil is energized. Upon energizing the latching coil, the mechanical latch is released, allowing the armature to open. Contacts are $\frac{3}{16}$ in. in diameter, rated 8 amp 115 v a-c, or 10 amp 24 v d-c, non-inductive. Arrangement of contacts is double pole double throw. Coils are available for any voltage from 6 to 115 v a-c, 50 to 60 cycles, and 2 to 32 v d-c. The relay is mounted on a bakelite base measuring $1\frac{7}{8}$ x $3\frac{3}{8}$ x $1\frac{3}{4}$ inches.

Leach Relay Co., 5915 Avalon Blvd., Los Angeles 3, Calif.



Circle No. 4 on Reader Service Card

Two Resins Insulate Electrical Machinery

Two silicone resins for electrical applications have been announced. One is a flexible resin of good drying speed. It is tough, fast-drying without a catalyst, has excellent build on glass cloth, and good heat life. It is intended for use as a coating on glass cloth and glass sleeving, and as a sticker for mica-glass tape and other flexible mica products. It is supplied as a 60 percent solution in toluol for application.

The other new resin is supplied as a 50 percent solution in petroleum spirits. It was designed for bonding and impregnating motors, generators, and transformers requiring Class H insulation. However, it may be used in bonded mica, mica tape, and glass combinations. When extreme flexibility is not desired it can be used effectively as a coating for glass cloth and sleeving. When properly cured it is a relatively hard resin but maintains a high degree of flexibility. It will through-cure readily and develop good bonding properties at elevated temperatures. It is to be used as furnished; no catalyst is necessary.

Chemical Dept. of General Electric Co., Pittsfield, Mass.

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CONTINUED ON PAGE 172

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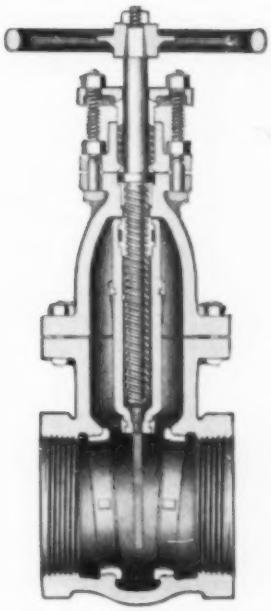
Designers' favorite for piping specs

...your CRANE Catalog



You can't find a more complete selection of piping materials listed anywhere. Not only listed, but carefully arranged and cross-indexed to save you the bother of having to hunt for the items you need. Just remember, if it's piping, chances are it's in the Crane Catalog... and easy to find.

Other piping procedures, too, are simplified when you make Crane your one source of supply. In your plant, for example, product assembly operations go smoother. In the field, replacements are easier to get.

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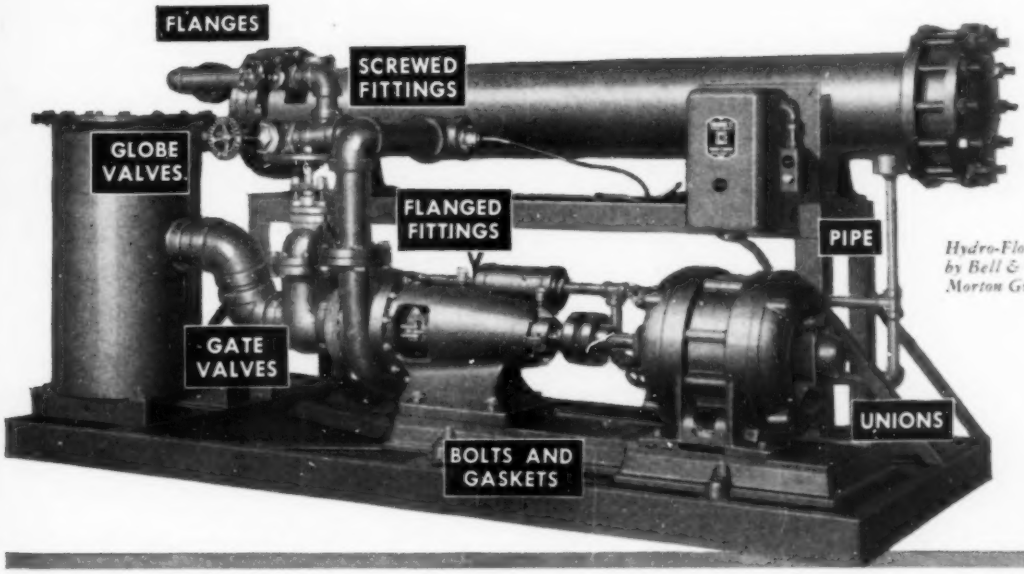
CRANE CO., 836 S. Michigan Ave., Chicago 5, Ill.
Branches and Wholesalers Serving All Industrial Areas

FOR STEAM, WATER, OR OIL *lines Crane offers a wide selection of 125-Pound Iron Body Gate Valves, brass trimmed or all-iron, in patterns for every need. Shown here, Crane No. 460 with non-rising stem, screwed-in brass body seat rings, brass disc faces. Working Pressures: 125 pounds steam; 200 pounds cold water, oil, or gas. Sizes: 2 to 8-inch. Screwed ends. See your No. 49 Crane Catalog, p. 77.*

● FOR COMPLETENESS OF SELECTION ... YOU CAN'T BEAT THE CRANE LINE



Hydro-Flo Oil Cooler, by Bell & Gossett Co., Morton Grove, Ill.

EVERYTHING FOR EVERY PIPING SYSTEM

CRANE

VALVES • FITTINGS • PIPE • PLUMBING AND HEATING

Fluid Coupling Gives Constant Linear Speed

This coupling was developed for roll winding applications requiring constant linear speed and tension in the material being wound. The hydraulic feature permits operation at stall under some conditions without damage to motor or coupling.

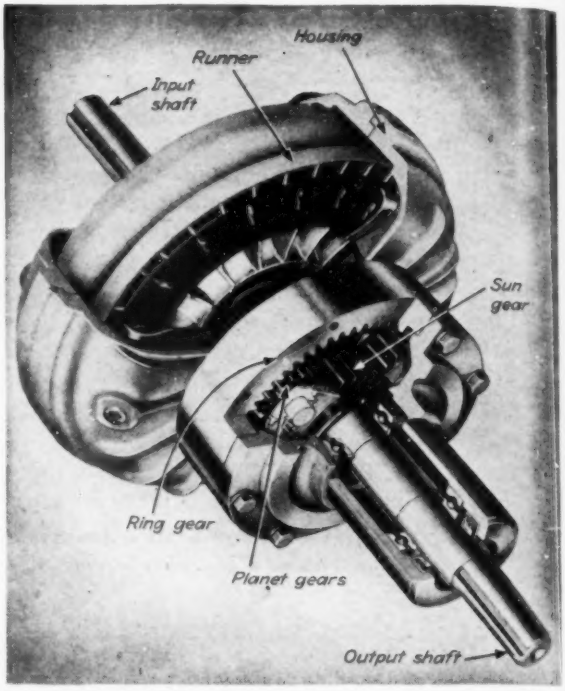
The ring gear of the planet set, the outer member or housing of the coupling, and the input shaft are all fastened together to form the input member of the unit. The planet gears through their carrier are keyed to the runner member of the hydraulic coupling. The sun gear is integral with the output shaft.

The input speed to the coupling is held relatively constant by powering the job with a motor of about 2 to 2½ times the required output hp. Running free, output speed equals input, with no torque transmitted. As torque is imposed, output-input speed ratio decreases. At either free running or complete stall the output hp is zero. Between these two conditions the output power increases to a maximum and then falls.

The balancing member is composed of the runner blades keyed to the planet carrier of the gear set. At no load, there is no relative motion between gears or between hydraulic runner and impeller. As load is applied the sun gear slows up, causing the planets to rotate on their own axis. The coupling acts as a clutch between these two members with a fixed rate of maximum slip, and the drag of the sun gear gives increased transmitted torque with decreased output speed.

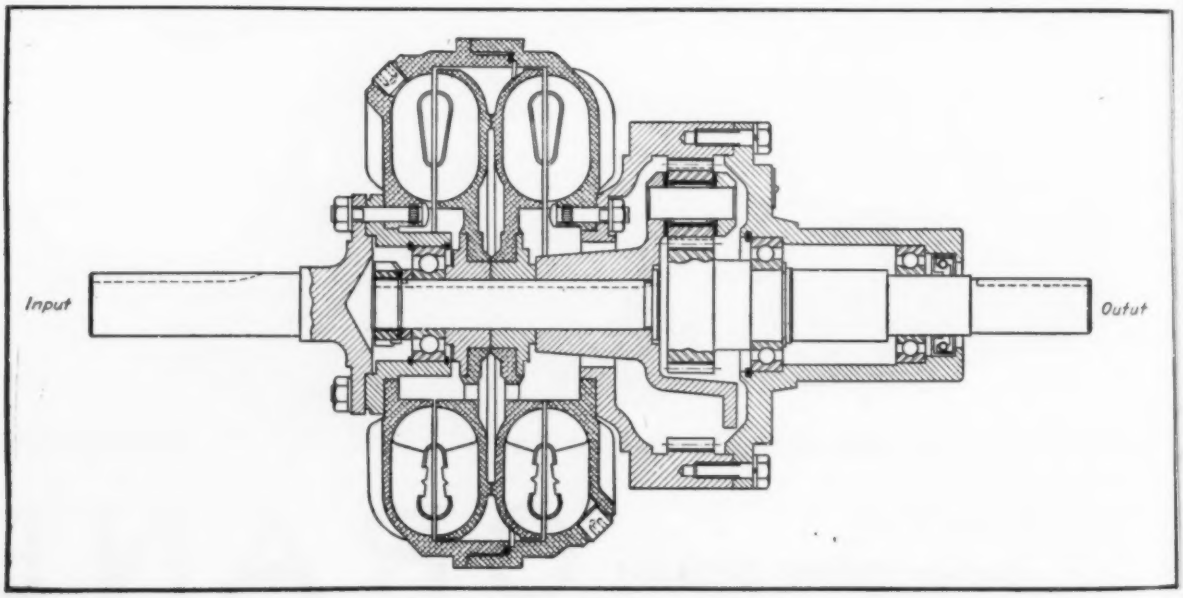
The ratio of the gears limits the coupling slip to 31 percent with the sun gear stalled. This slip control keeps the unit within practical heat dissipating limits.

The couplings measure 16½ in. long by 9¾ in. diameter. Two capacities are available for inputs from 500 to 1,700 rpm.



PLANETARY GEAR SET is combined with hydraulic coupling to give near-constant power output with steady input speed. A motor of approximately twice the required output hp is used to keep input speed relatively constant. Output power rises from zero to a maximum and then returns to zero in a parabola-like curve as the output speed decreases. The couplings measure 16½ in. long by 9¾ in. diameter. Two capacities are available for inputs from 500 to 1,700 rpm. Original application was in paper and textile winding machines.

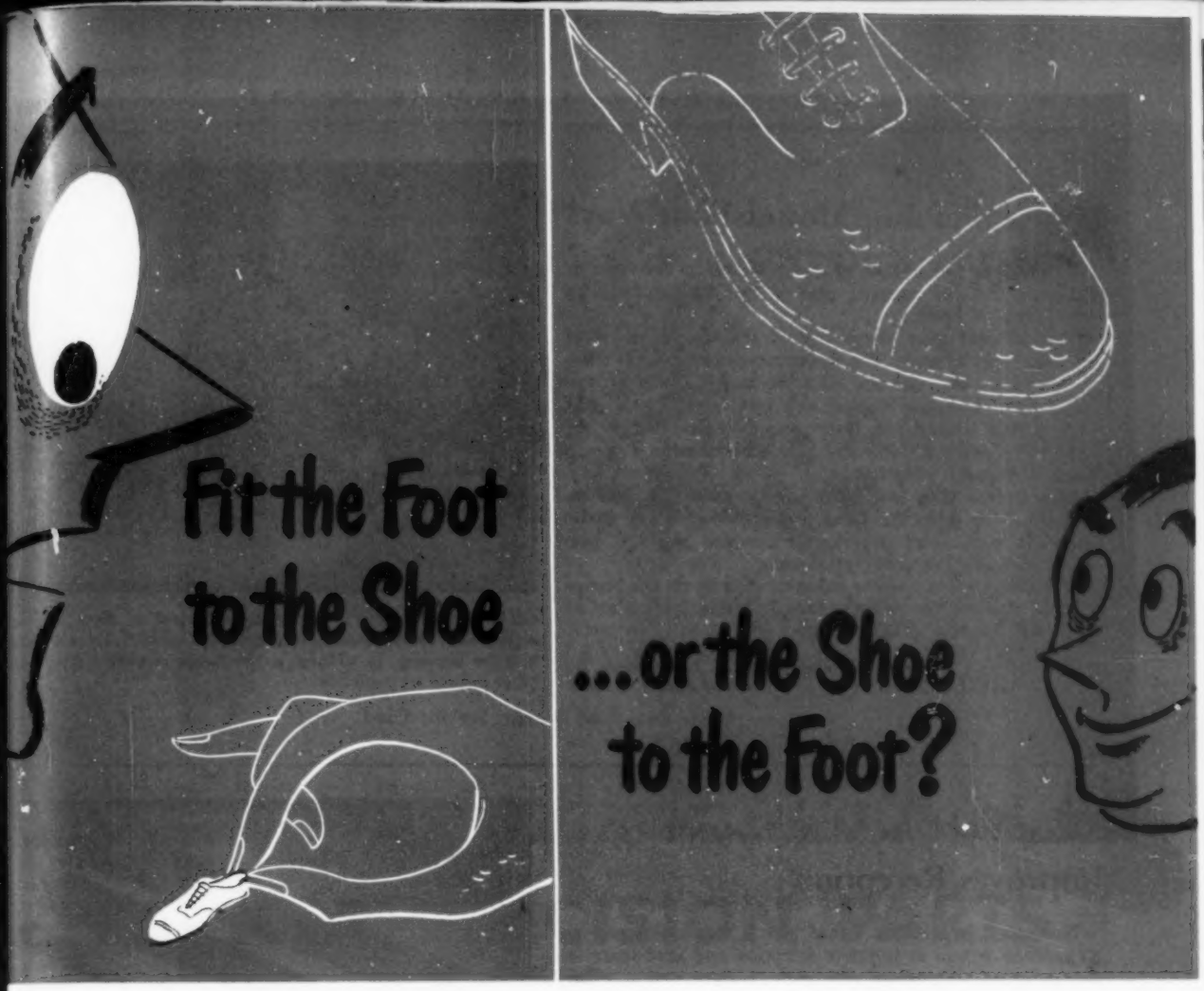
Twin Disc Clutch Co., Racine, Wis.



Circle No. 6 on Reader Service Card

CONTINUED ON PAGE 174

UNIVERSITY OF MICHIGAN LIBRARIES



In *friction materials*, R/M can *always* fit the product to the need . . . because R/M makes 'em all!

When you entrust your brake and clutch problems to R/M, there's no chance of getting a "misfit" material. R/M, the largest producer in this field, makes every known type of friction material . . . and is constantly developing new types for new and different applications. Thus your R/M representative has no axe to grind for any one kind of material. Whatever your needs, he can recommend the right friction product . . . and he can call on the productive capacity of four great plants to make deliveries that suit your schedule.

RAYBESTOS-MANHATTAN, INC.

EQUIPMENT SALES DIVISION **620 Fisher Bldg., Detroit 2, Mich.**

345 Lake Shore Drive, Chicago 11, Ill. 4651 Pacific Blvd., Los Angeles 11, Calif.

1071 Union Commerce Bldg., Cleveland 14, Ohio

Factories: Bridgeport, Conn. Manheim, Pa. Passaic, N.J. No. Charleston, S.C.

RAYBESTOS-MANHATTAN, INC., Manufacturers of Brake Linings • Brake Blocks • Clutch Facings Fan Belts • Radiator Hose • Mechanical Rubber Products • Rubber Covered Equipment • Packings Asbestos Textiles • Powdered Metal Products • Abrasive and Diamond Wheels • Bowling Balls

R/M

FIRST IN FRICTION

Hydraulic Pump Mounts Four Ways

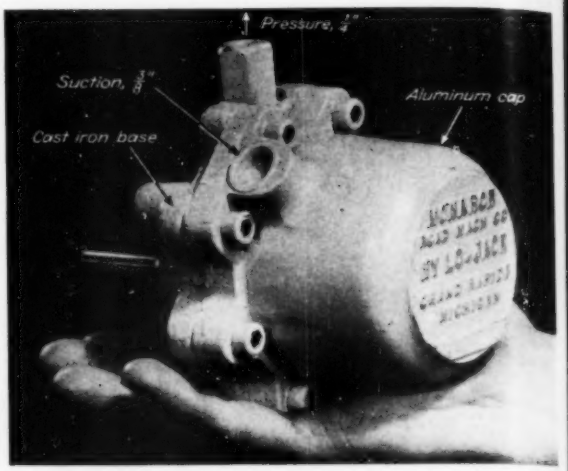
One gallon per minute of hydraulic oil is delivered at 1,000 psi by this compact gear pump. Less than 1 hp is required for operation.

The internal gear pump unit is replaceable in all of the four types offered. The flange mounted model is illustrated. Also available are foot mountings, pumps combined with three-way valves, and complete control units with pumps, valves and reservoirs. Needle bearings are provided at contact points and pumps may be run at motor speed or through reductions.

Helical type alloy steel gears are the pumping mechanisms. Shafting is hardened and ground. The filtering screen is double mesh brass and can be removed together with the relief valve assembly, for cleaning.

Three way valves may be mounted directly on the pumps or kept separate for remote control. Valve bodies are of steel and handles of aluminum.

Monarch Road Machinery Co., 327-29 Front St., Grand Rapids 4, Mich.



GEAR PUMP weighs 7 lb complete and is 7 in. long. A filter screen and compression spring pressure relief valve are built into the housing. In addition to the flange mounting type, four other casings are available.

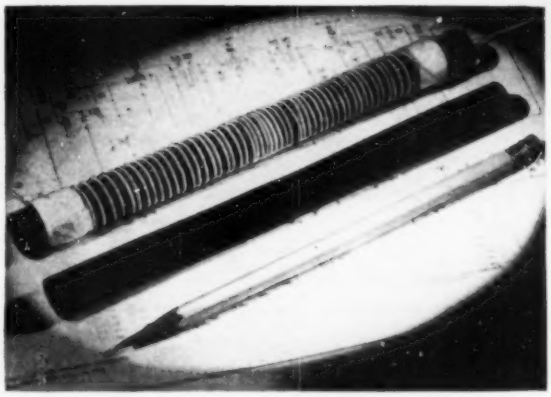
Circle No. 7 on Reader Service Card

Metal Core Antenna System Improves Reception

Not much larger than a lead pencil, this coil serves as an antenna system to improve reception and save space in the radio set. Signal-to-noise ratio is increased, thereby reducing man-made static or background noise.

The new rod or tube antennas are produced from 0.250 to 0.6 in. in diameter, but present facilities can turn out up to 1½ in. diameters. Square rods and tubing, or fluted cross-sectional shapes for minimum winding contact can be furnished. The material is manufactured in various lengths with cores averaging 7 to 8 in. long at present.

With a single layer winding of insulated wire, the assembly is used in place of the conventional outside antenna or the built-in loop. Signal pickup is considerably better than that of the usual air loop, on the order of 2:1 at the low end of the broadcast band, and about equal at the high. Q's above 200 are obtainable.



ANTENNA SYSTEM comes in compact rod or tube assembly, replacing outside antennas and built-in loops. Static and noise exclusion is claimed superior with the new component.

Henry L. Crowley & Co., Inc., 1 Central Ave., West Orange, N. J.

Circle No. 8 on Reader Service Card

Thread Bulge Locks Screw Against Vibration

This screw is designed to lock in any position without external devices. Vibration has no effect; the screw is removable only by continuous reverse torque. It can be removed a number of times without injury to the threads.

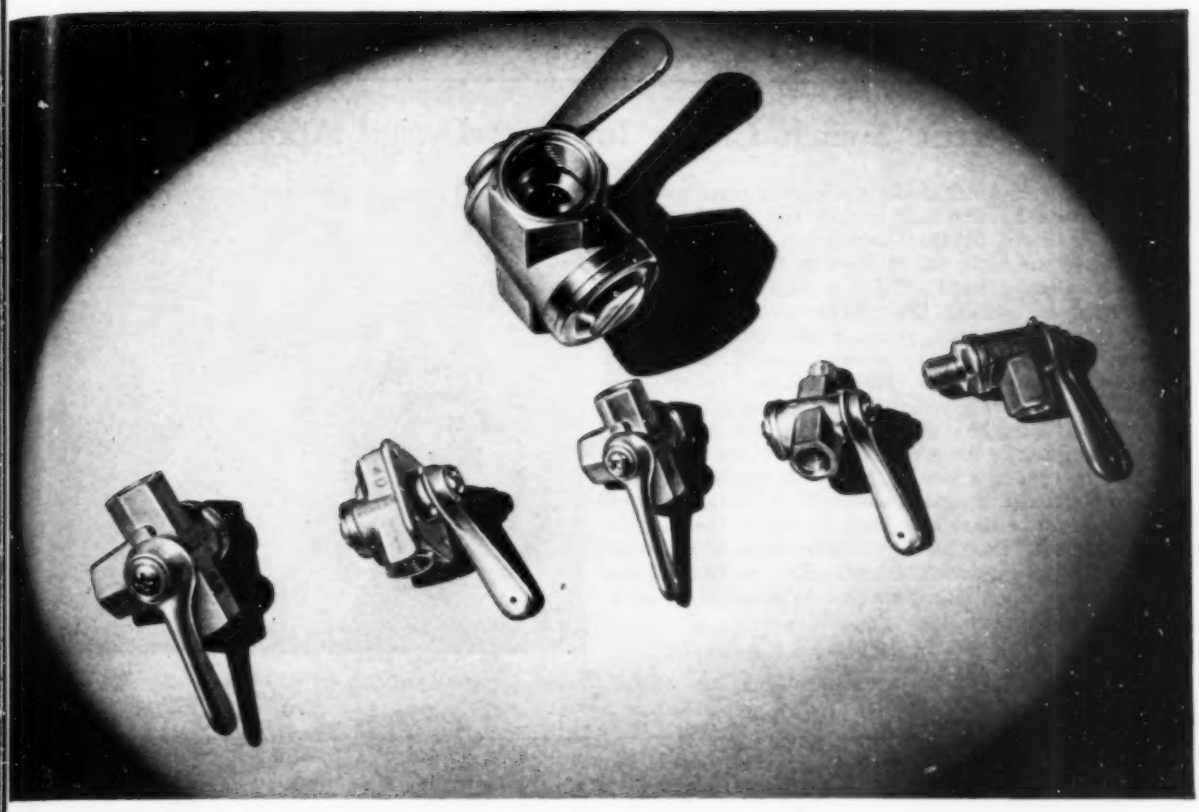
The principle is simple. At the point where the thread locking takes place, a portion of the screw thread is removed. The thread on one side of this space has a gently curved bulge which projects into this vacant area. Since the path through which the mating thread of the workpiece must pass is no longer straight, the mating thread re-forms to conform to the bulge thus providing a positive locking action internally. This locking action can be placed at any location specified.

Thredlock, 9397 Whitall Lane, Grosse Ile, Mich.

Circle No. 9 on Reader Service Card

CONTINUED ON PAGE 176

UNIVERSITY OF MICHIGAN LIBRARIES



KOHLER PRECISION PARTS
FOR
Aircraft, Industrial, Automotive Applications

The complete Kohler line includes a wide variety of precision valves, fittings and other units, manufactured to customers' specifications. The plug valves illustrated are used for the control of oxygen, air, water, alcohol, oil, gasoline and other fluids. Kohler plug valves have forged aluminum bodies with precisely-lapped steel plugs. Connections that best suit your installation can be provided. Leakage is prevented at three vital points: handle, bottom cap and around the plug.

With an organization thoroughly experienced and skilled in precision workmanship, Kohler maintains full facilities for forging, tooling, machining, finishing, assembling, testing and inspecting. Prompt, accurate deliveries can be made.

Kohler engineers will develop valves or fittings for volume production to meet special requirements. Write for a complete catalog and send your specifications or requirements to Kohler Co., Dept. 3-V, Kohler, Wisconsin. Established 1873.

KOHLER OF KOHLER

PLUMBING FIXTURES • HEATING EQUIPMENT • ELECTRIC PLANTS • AIR-COOLED ENGINES

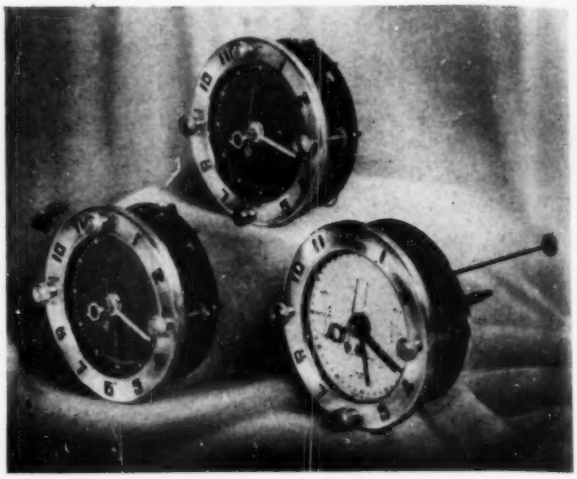
Electric Switch Clocks Improve Design Appearance

A new line of switch clocks is offered to provide designers with a combination built-in clock and period sequence timer. Approved by the Underwriters, all of the new mechanisms are fused for 15 amps at 125 volts. Control knobs may be furnished in front or back and a variety of clock hands are available. Dials can be styled to specification.

An "on-off" automatic switch is used to open or close the circuit manually or to throw operation over to the electric brain of the clock. No other cutout switch is required. With other adjusting buttons the on and off timer for the switch can be set. The clock runs continually.

The unit may be flange mounted in a counterbored hole or may be held in a straight hole with special mounting rings. A number of models are available. Adjustment of the hands can be done by a knob located on the front or back. Extended back hand adjustment knob is shown at bottom right in illustration. A knob setting switch-on operation for some future time is optional, as shown in the other two models illustrated.

The Sessions Clock Co., Forestville, Conn.



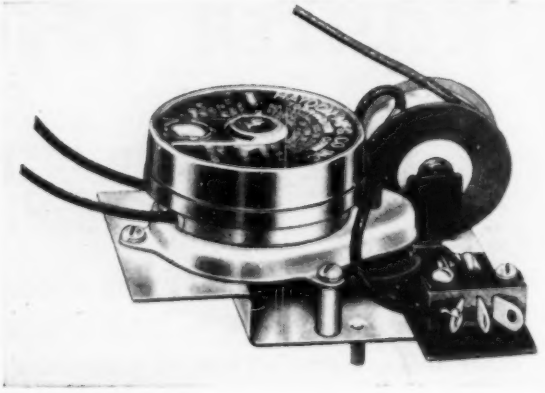
Circle No. 10 on Reader Service Card

Mass Production Electric Timer Gives Audible Signal

A low-cost timing device has been offered with an audible signal for large volume applications. A range of timing intervals is available. At the end of the preset interval the signal, a buzzer, sounds continuously until manually turned off, demanding whatever action is required.

The unit, measures $4\frac{1}{32}$ by $2\frac{11}{32}$ by $1\frac{25}{32}$ in. It can be mounted and operated in any position; rotates clockwise or counter-clockwise and is adjusted in either direction, to start or when in operation. With a snap action break, the unit is rated for 16 amps 250 v. $\frac{1}{2}$ hp 250 v a-c. The basic motor operating the device features a 450 rpm rotor speed. The motor is totally enclosed, with a dual system of controlled lubrication. According to the manufacturer, the timer lends itself to economical volume production and is not available in small quantities.

Haydon Manufacturing Co., Inc., Torrington, Conn.



Circle No. 11 on Reader Service Card

Packing Rings Withstand Chemicals At High Temperatures

Where applications demand resistance to chemical attack and free spindle operation, a new cone type packing can be used in stuffing boxes. The seals are made of Teflon, the new DuPont material, and are claimed to be inert to all chemical action. Working temperatures from —150 to 550 F are permissable. Valve "freezing" is prevented by the natural slipperiness of the packing material.

Cone-shaped ring packings are supplied. Sealing action is derived partly from the flexing of these cones and does not depend entirely on deformation. Less pressure is required for flexing than for deformation, and low gland pressures exert enough of a flattening action to cause the seal to bear on the stuffing box and the spindle and form an effective seal.

Packing is available with adapters for square- or chamfered-end boxes. The rings may also be split for installation.

U. S. Gasket Co., P. O. Box 93, Camden, N. J.

Circle No. 12 on Reader Service Card

CONTINUED ON PAGE 178

UNIVERSITY OF MICHIGAN LIBRARIES

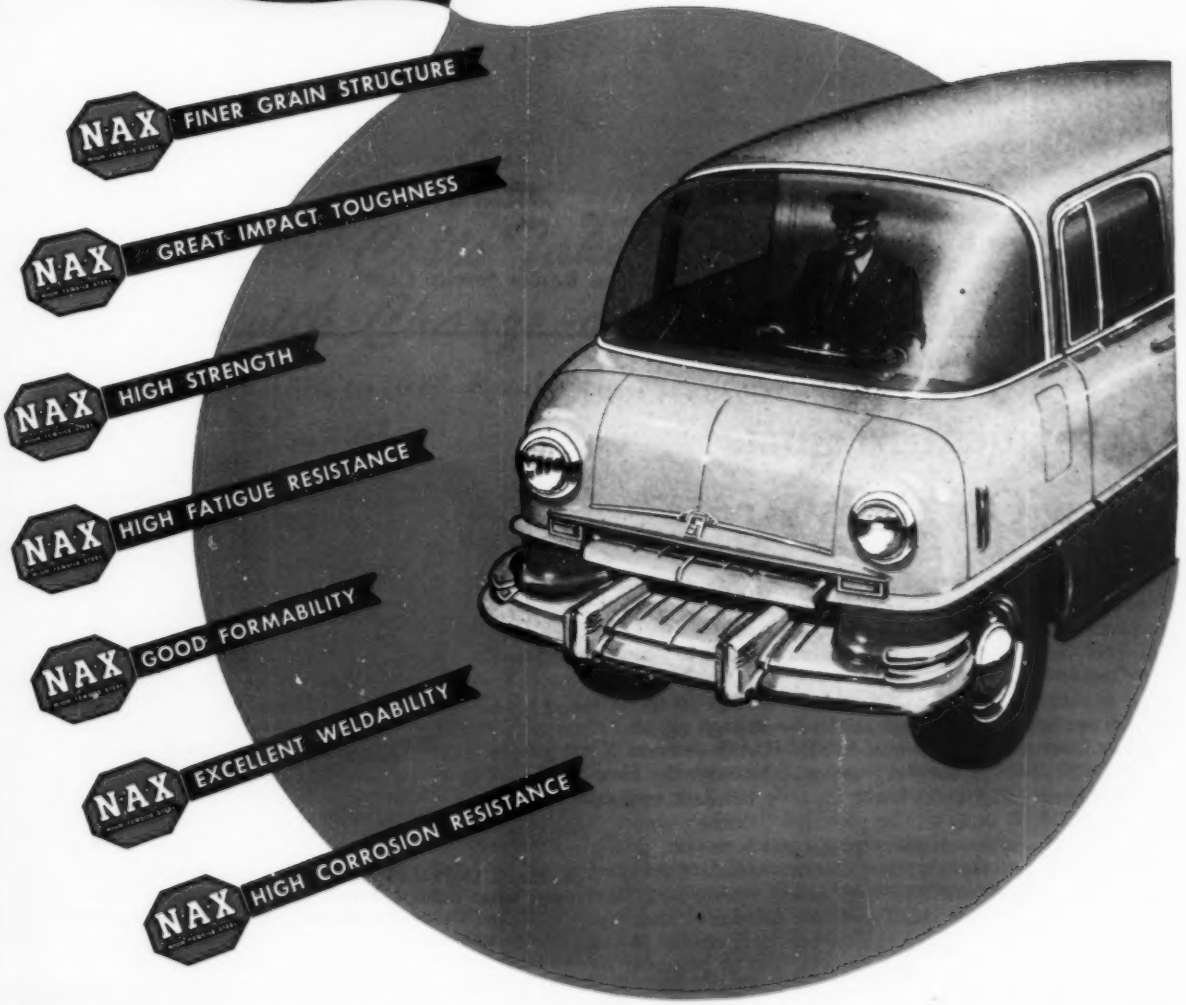
MAKE A TON OF SHEET STEEL GO FARTHER

Specify—

N·A·X

HIGH-TENSILE STEEL

SEVEN STRONG REASONS explain the trend to N·A·X HIGH-TENSILE steel in the manufacture of commercial vehicles.



GREAT LAKES STEEL CORPORATION

N-A-X ALLOY DIVISION • ECORSE, DETROIT 29, MICH. • UNIT OF NATIONAL STEEL CORPORATION

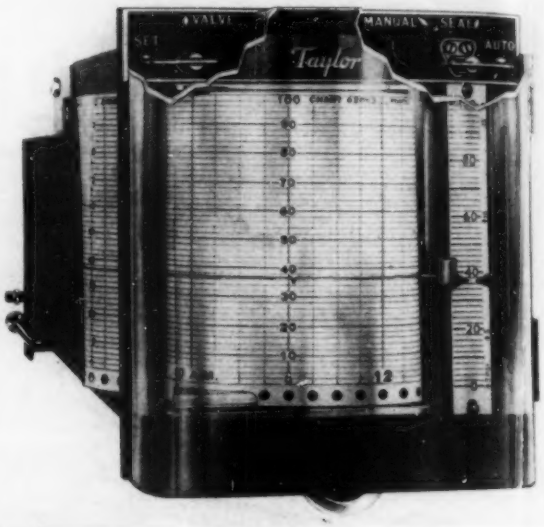
Graphic Recorder Saves Panel Room

This miniature recording-receiving instrument is designed for pneumatic transmission of flow, liquid level, pressure and temperature data. It fits into a panel opening of $3\frac{7}{8}$ x $4\frac{1}{2}$ in. with a flange measurement of $4\frac{3}{8}$ x 5 in.

The door of the recorder is made of clear plastic to give full view of the chart, scale and principal instrument settings. When the door is opened, adjustments can be made for checking of the valve position and for automatic-manual service. Set-point adjustment is made by turning the knurled knob at the bottom of the case. Two micrometer screws above the automatic-manual lever provide a convenient means for making zero adjustments for pen and set-pointer.

The transmitter output is recorded in terms of the process variable measured. The 3 in. wide chart travels from right to left at 1 in. per hour. Charts may be torn off from left side as needed. Electrically driven chart mechanism is designed to pull out like a camera. The scale at the right shows actual process range of the transmitter and is graduated 3 to 15 psi for noting valve position and checking calibration.

Four forms (all receivers 3 to 15 psi) are available: (1) one process record, set-point indication, set-point adjustment, valve position indication and automatic-manual unit for use with controllers located in the control house; (2) one process record, set-point indication, set-point adjustment, valve position indication and automatic-manual unit for use with field mounted controllers; (3) one process record, valve position indication and remote manual valve loading; (4) one process record only.



GRAPHIC PANELS can be made more compact by using this pneumatic receiver-recorder to give record of processing data right at the point in the panel at which it is required. Flange measures $4\frac{3}{8}$ by 5 in.

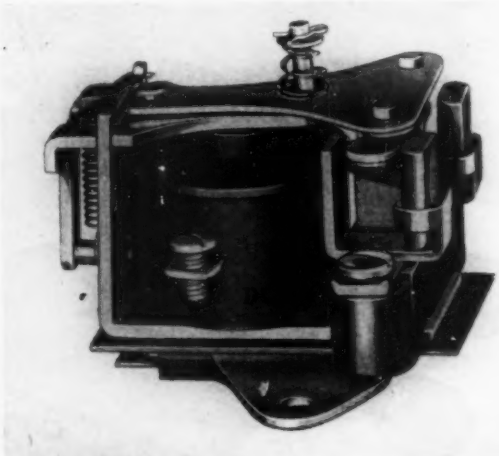
Taylor Instrument Co., Rochester 1, N. Y.

Circle No. 13 on Reader Service Card

Permanent Magnets Blow Out Relay Arcs

By the incorporation of two permanent magnets to blow out the arc, the carrying capacity of this d-c relay has been significantly increased. The arc is sucked away from the current carrying elements and contact burning is reduced.

Contact load in excess of 5,000 w or 20 amp at 230 v d-c is claimed to be possible without damage to contacts. Coils are available for any a-c or d-c voltage up to 230 volts. Contact arrangement of SPST/NO-DB (1 form X) and DPST/NO (2 form A) are available with the blow-out magnets. However, lower capacity auxiliary contacts in forms, A, B or C can be provided. Contact carrier is full floating, with a clean wiping contact motion, giving heavy contact pressure without chatter. Coils are designed for continuous operation. The return spring has been liberally designed for quick armature movement and long spring life. The operating coil carries triple enamel coated copper magnet wire, random wound, and is protected by thoroughly impregnated baked varnish.



THE FULL-FLOATING CONTACT arm carries 5/16 in. diameter silver contacts. Metal parts are plated for corrosion resistance, and the bobbin and terminal board are molded bakelite. Multi-strand copper pigtails carry current across the hinge. Two alnico permanent magnets blow out the arc.

Potter & Brumfield, Princeton, Ind.

Circle No. 14 on Reader Service Card

CONTINUED ON PAGE 180

UNIVERSITY OF MICHIGAN LIBRARIES

Assembly Costs Take a Tumble with Change-over to CLUTCH HEADS

Users Certify 15% to 50% Production Increases for Lower Final Costs

In view of this testimony you may confidently expect these exclusive time and cost-saving CLUTCH HEAD features to deliver a similar ratio of production increases on your assembly line . . . to lower your final costs by producing more for less.

Higher Visibility of the clutch recess eliminates operator hesitation. Even "green" help drives with speed and confidence. No "break-in" needed.

No Damaged Heads . . . Dead-center entry with the Center Pivot Column prevents driver canting, makes straight driving automatic, and checks out delay and expense fixing burred or chewed-up heads.

Now, Non-Tapered Driving that sends skid damage to zero; safeguarding manpower and material. With CLUTCH HEAD'S all-square driving engagement there is no need for end pressure to combat "ride-out" (as set up by tapered driving) and the drive home is effortless. This safety factor and elimination of fatigue steps up production.

One-Handed Reaching at "Bottlenecks." Only CLUTCH HEAD provides a frictional Lock-On that joins screw and bit as a unit to permit one-handed reaching into inner spots and driving from any angle.

214,000 Screws Driven Non-Stop. This is the record established by the rugged Type "A" Assembly Bit . . . continuous high torque driving on a main assembly line of one of America's largest automotive plants.

New Bit Life in 60 Seconds. Consider the added tool economy of simplified reconditioning this bit REPEATEDLY . . . by a 60-second application of the end surface to a grinding wheel.

Curing Field Service "Headaches." For simplified field service, CLUTCH HEAD alone has a recess that is basically designed for operation with a common screwdriver or with any flat blade which need only be reasonably accurate in width.

These advantages are fully detailed in the New CLUTCH HEAD Brochure . . . along with technical information your engineers and plant executives will want for reference. Start your investigation of CLUTCH HEAD'S potential savings for your assembly by sending for a copy . . . and indicate sizes and types of screws which interest you.

UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 · CHICAGO 8 · NEW YORK 7

Compact Casting Houses Air Regulator Unit

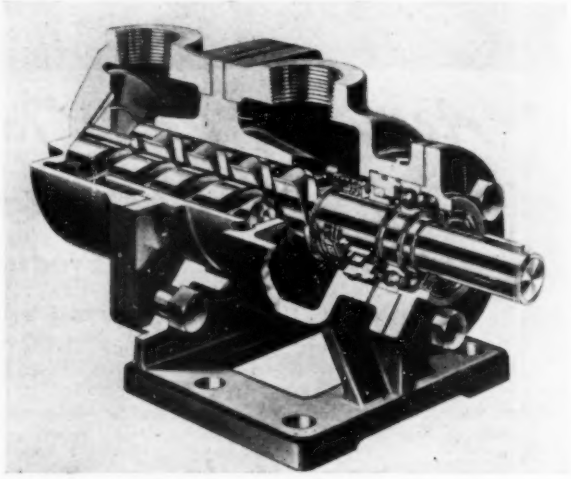
Connected in the air intake line, this air regulator-filter-lubricator has no projecting parts and can be mounted on a central control panel. It incorporates a piston-type reducing valve, a pressure gage, a lubricator and a synthetic-type air filter. Incoming air is pressure regulated, filtered and provided with lubricating oil for the protection of working parts in the system.

The filter cover and regulating valve screw directly into a 6 x 4¾ in. body casting. Two front panel knobs regulate air pressure and the amount of oil that is injected. A third knob control drains the water trap. An air pressure gage and an oil level indicator are also provided. All operating units are enclosed in the cast body and no glass parts or fragil mechanisms are exposed to breakage. Filter is replaceable from the front of the unit through the removable cover.

Logansport Machine Co., Logansport, Ind.

Circle No. 15 on Reader Service Card

Mass Production Pump Has Three Working Parts



A pulsation-free, high speed pump feeds petroleum products and other light or viscous fluids in volumes up to 85 gpm against pressures up to 150 psi for oil burners, oil transfer, lubrication or hydraulic systems. The new pump is a standard, mass-production unit.

There are three working parts. A power rotor meshes with two sealing or idler rotors in such a manner that the liquid is conveyed continuously from inlet to outlet. Direct-connected electric motors or turbines, without reduction gears, belts or chains furnish the drive.

All parts are interchangeable. The mechanical seals require no attention. Suction can be located at top, bottom, horizontal left or horizontal right, and the pump can be mounted in any position.

DeLavel Steam Turbine Co., Trenton 2, N. J.

Circle No. 16 on Reader Service Card

Combination Timing Switch Covers Wide Range

This switch is an automatic timer, recording intervals from $\frac{1}{20}$ sec to 4 min, with an accuracy within 2 percent. Combined in it are four basic types of timing: interval, delayed action, automatic repeat and programming. The type of operation is determined by external connections to the terminal board. A number of timers can be connected together for more complex programming.

The switch employs only one tube and operates from the discharge of a timing capacitor in the grid circuit. Period control is accomplished by a selector switch with ranges of 1 sec, 10 sec, 1 min and 4 min. A variable resistor provides for intermediate time intervals between fixed selector settings.

Standard units are supplied for operation on either 115 v or 230 v, 50-60 cycles. All relay contacts are rated at 10 amp to 115 v, a-c, 5 amp to 230 v a-c, non-inductive load.

Coral Designs Div., Henry G. Dietz Co., Box 248, Forest Hills, N. Y.

Circle No. 17 on Reader Service Card

CONTINUED ON PAGE 182

UNIVERSITY OF MICHIGAN LIBRARIES

Words That Sell

MADE OF STAINLESS STEEL

For the Most in Sales Appeal . . Begin with Sharon Stainless Steel.

Products **Made of Stainless Steel** just naturally have better acceptance than products made of other materials. The clean, sanitary appearance — the rust and corrosion resistant qualities — the lifetime finish that never loses its brightness — the easy-to-care-for characteristics of products **Made of Stainless Steel** makes them better products — better selling products.

Fabricators: Stainless steel is stronger — can be used in lighter gauges without sacrificing strength. It is easy to work — existing machines are usually satisfactory. Plating and other operations eliminated — Stainless frequently costs less to use.

For product information and fabrication data check with Sharon — Pioneer and prime producers of Stainless Steels. Offices listed below and in yellow pages of phone directories.

SHARONSTEEL

SHARON STEEL CORPORATION *Sharon, Pennsylvania*

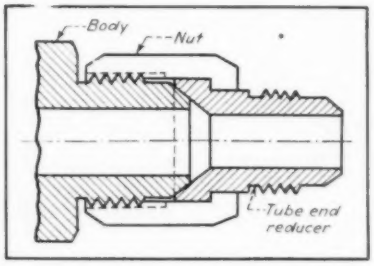
PRODUCTS OF SHARON STEEL CORPORATION AND SUBSIDIARIES: THE NILES ROLLING MILL COMPANY, NILES, OHIO; DETROIT TUBE AND STEEL DIVISION, DETROIT, MICH.; BRADLEY STEEL COMPANY, WARREN, OHIO; SHARONSTEEL PRODUCTS COMPANY, DETROIT, MICHIGAN, AND FARRELL, PENNA.; CARPENTERTOWN COAL & COKE CO., MT. PLEASANT, PENNA.; FAIRMONT COKE WORKS, FAIRMONT, W. VA.; MORGANTOWN COKE WORKS, MORGANTOWN, W. VA.; JOANNE COAL COMPANY, RACHEL, W. VA. Hot and Cold Rolled Stainless Strip Steel—Alloy Strip Steel—High Carbon Strip Steel—Galvanic Special Coated Products—Cooperage Hoops—Hot Steel Sheets—Hot Rolled Annealed and Descaled Sheets—Galvanized Sheets—Enameling Grade Steel—Welded Tubing—Galvanized and Fabricated Steel Strip—Steel Strapping, Tools and Accessories.

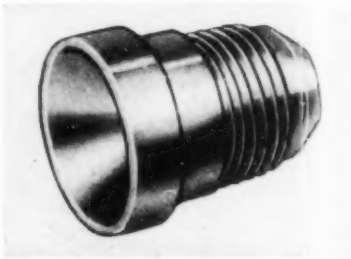
DIRECT SALES OFFICES: Chicago, Ill., Cincinnati, O., Cleveland, O., Dayton, O., Detroit, Mich., Minneapolis, Ind., Milwaukee, Wis., New York, N. Y., Philadelphia, Penna., Rochester, N. Y., Los Angeles, Calif., San Francisco, Calif., St. Louis, Mo., Montreal, Que., Toronto, Ont.

Reducer Steps Down Tube Size with Standard Fittings

A new tube end reducer for use with the manufacturer's standard fittings connects different size tubings. Any standard fitting can be converted to a reducing type. Attached with a standard nut to the fitting tube end, the reducer provides a flared nose for the smaller tube size and threads for the smaller nut.

In steel or brass, the new fittings are available in 8 sizes: $\frac{3}{8}$, $\frac{1}{2}$ and $\frac{3}{4}$ to $\frac{1}{4}$ or $\frac{3}{8}$ in., $\frac{3}{4}$ to $\frac{1}{2}$ in. and 1 to $\frac{3}{4}$ in. tube.





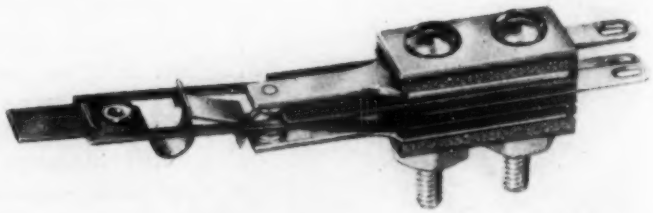
Parker Appliance Co., 17325 Euclid Ave., Cleveland, Ohio

Circle No. 18 on Reader Service Card

Snap Switch Handles High Inductive Load Currents

A snap-action switch for use with relays, limit switches and other applications is now being manufactured. Ampere rating is high and the switch is designed to take high inductive loads with a minimum of arcing. For certain applications, the switch is adjustable to operating movement and over-travel.

The main operating blade is made of tempered spring steel and can be furnished to operate at various pressures. Conventional switch can be used for singel pole, single throw, or single pole, double throw requirements. Special types available on specification. Unit is compact in size and weighs about one ounce.



Comar Electric Co., 3148 N. Washtenaw Ave., Chicago 18, Ill.

Circle No. 19 on Reader Service Card

Pressure Indicator Takes Vibrations and Heat

These three new pressure indicators are used in applications where high frequency vibration or heat would cause inaccuracies if other measuring devices were used. Maximum dynamic pressures are 500, 1,000, 2,000 and 3000 psi for the indicators now offered. Static pressures are 50 percent higher in each case.

These indicators are capable of reproducing pressure fluctuations in an engine cylinder with an accuracy of 1 percent over the entire range of the pick-up, thus making it possible to investigate ignition phenomena, pressure-power functions and to accumulate other data hitherto impossible to measure accurately. The high natural frequency of the pick-up (approximately 45,000 cps) gives it wide frequency response range, which extends from static pressures to frequencies over 20,000 cps.

For engine analysis, the pressure pick-up units are normally used with an amplifier power supply unit, which receives the output signal from the pickup, amplifies and supplies a signal suitable for feeding a standard oscilliscope for graphical presentation of the pressure-time curve. Calibration of the scope trace is obtained by actuating a panel switch on the amplifier, which provides standard reference voltages corresponding to known values of pressure as read on a calibrated dial. The amplifier is battery-operated and entirely self-contained, including space for the pick-up and the cable when not in use.

All of the pressure pick-ups listed above utilize a catenary diaphragm of negligible inertia actuating a tubular strain generating tube on which strain gage wires for force measurements are wound in such a way as to minimize inductance and make the units temperature compensated. High frequency vibrations are taken without damage.

Control Engineering Corp., Canton, Mass.

Circle No. 20 on Reader Service Card

CONTINUED ON PAGE 184

UNIVERSITY OF MICHIGAN LIBRARIES

"Commercial" Ball-Bearing Data...#9 of a Series

WE WILL BE GLAD TO SEND YOU COPIES OF THE PREVIOUS ADVERTISEMENTS IN THIS SERIES.

How Schatz "Commercial" Ball Bearings can cut Anti-Friction Costs 35 to 85%

These tables show how Schatz "Commercial" Ball Bearings can deliver required anti-friction performance at ⅔ to ⅙ the cost of expensive precision bearings.

Take, for example, an application in which the radial load is 290 lb., the thrust load is 40 lb., and shaft speed is 150 rpm.

Equivalent radial load is 330 lb. (290+40).

From the speed-load curve in Fig. 1, the correction factor at 150 rpm is 1.65. Dividing 330 lb. by 1.65 gives 200 lb.—the loading at 600 rpm.

Fig. 2 shows two bearings which will support this load. One has 13 balls, 5/16" in diameter, and the other has 20 balls of ¼" diameter.

Fig. 3 gives minimum O.D. and maximum bore for these two bearings. For that of thirteen 5/16" balls, O.D. is 2", bore is 13/16". For that of twenty ¼" balls, O.D. is 2¼", bore is 1¼".

If Ground "Commercials" are used, the load-carrying capacity is increased 2.17 times. To use the tables, it is necessary only to divide the computed equivalent radial load by 2.17.

If thrust load is greater than radial load, a radial-thrust or straight thrust bearing should be used.

It will pay you to investigate the savings possible with "Commercials" where operating conditions are within the load and speed limits of these versatile low-cost ball bearings. Or if your application requires it, our engineers can adapt or design a special Schatz Ball Bearing to fit your specific job requirements—often at less cost than a standard precision bearing.

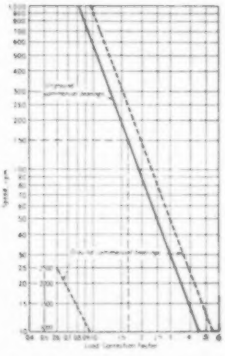


Fig. 1. Speed-load correction factors for "Commercials." This is necessary since bearing life is rated at 600 rpm.

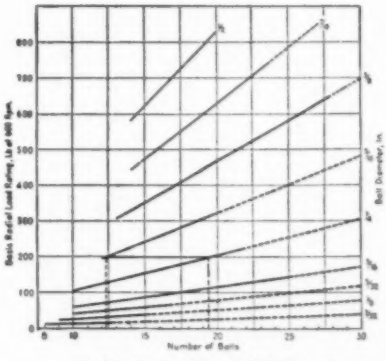


Fig. 2. Load rating at 600 rpm of ball complements used in Unground "Commercials."

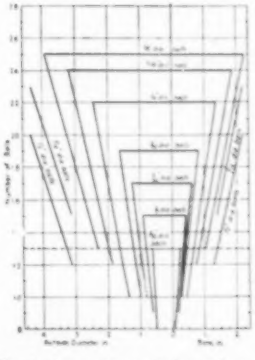


Fig. 3. Approximate maximum bore and minimum O.D. of Unground "Commercials."

THESE AND OTHER TABLES showing correct size and type "Commercial" Ball Bearings for various speed and loading conditions, and graphs for estimating bearing life, are included in "Construction and Characteristics of Low-Cost Ball Bearings." This technical booklet also discusses modifications that can be made for special design, assembly and operating conditions. Catalog No. 11 gives data on the complete Schatz "Commercial" line. Write for your copies today.

SCHATZ "Commercial" BALL BEARINGS

The Schatz Manufacturing Company
6758 FAIRVIEW AVENUE, POUGHKEEPSIE, N. Y.

Four Strain Functions Recorded Simultaneously

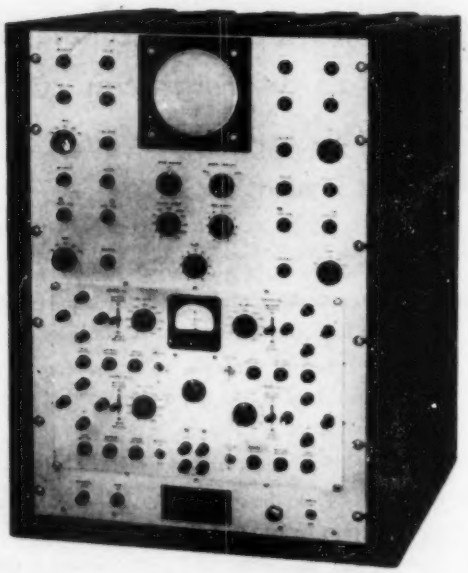
A new strain analyzer makes possible the simultaneous observation and recording of electronic traces of four strain functions in correct time relationship without the necessity of optical alignment. It can be used in studies of strain, impact, vibration and torque.

The instrument is designed for d-c bridge excitation from self-contained batteries and has high-gain amplifiers, balance controls, and voltage and calibration controls for each strain gage bridge channel. It can be used with strain gages of from 60 to 500 ohms resistance. There are four units; the indicator, the indicator power supply; the camera and the camera speed control. The indicator is housed in a steel cabinet 30 x 24 x 23 in. and is shock mounted to protect it from vibration and impact. The cathode ray tube is a special four-gun type. In designing the indicator cabinet and in placing the tube, provision has been made for installation and removal of a recording camera.

The strain gage bridge control panel is a removable plug-in unit. The bridge excitation batteries are contained within the indicator proper and may be replaced by external heavy duty batteries for long-term tests. Connection to the strain gages is made by means of four binding post terminals provided for each bridge, making all four arms available.

The calibration is in terms of one "active" and one "compensating" strain gage. When used in this manner, dummy resistors are connected across the other two arms. Correction may be made to permit the use of two or four active strain gages.

When using a single active gage with a gage factor of 2.0, the equivalent strain is 2,000 micro-inches per in. Multi-turn balance controls are provided for adequate balance range. A bridge voltage control is provided to allow for various strain gage resistances and to vary the sensitivity of each channel.



THE INDICATOR UNIT weighs 100 lb and stands 30 in. high. Auxiliary equipment includes a camera, electronic control, a periscope and mount assembly, and a power supply taking 300 w at 105–125 v, 50–60 cycle. Special cathode ray tubes are used for photographic applications. Four strain gage functions can be observed and recorded simultaneously in correct time relationship.

Baldwin Locomotive Works, Testing Equipment Dept., Philadelphia 42, Pa.

Circle No. 21 on Reader Service Card

New Phenolic Plastics Have High Impact Strength

Three phenolic molding compounds have been placed on the market. All three fall into the impact type classification and provide new properties or production performance. Two are high-impact, fabric-filled materials, while the third is a nitrile rubber-bearing, wood-flour-filled material.

The two new fabric-filled materials utilize two-step resins, and have improved flow properties and finish and are less critical when electronically preheated. One has an impact strength of 1.4 ft-lb per in. on the Izod scale. Its unusually good flow properties allow it to be either compression or plunger molded. It possesses a balance of regular mechanical properties and good electrical properties for this type material. The second fabric filled compound is similar but has almost double the impact strength (2.2 ft-lb per in.). Molding properties are not appreciably affected.

The third new material has the moldability of general-purpose materials, but because of its nitrile rubber content, it possesses improved impact strength and shock resistance. Its low modulus of elasticity (0.6×10^6 lb per sq in. in tension) allows it to be used in thin sections around metal insets. Molded parts are capable of standing repeated abuse without failure. The material can be plunger or compression molded and has good surface appearance. All three phenolics are characterized by high impact resistance. The fabric filled compounds have improved production properties and can be molded easily. Impact strength represents the only difference between the two. The nitrile rubber-bearing plastic, also impact resistant, is important for durability and ease of production.

Durex Plastics & Chemicals, Inc., No. Tonawanda, N. Y.

Circle No. 22 on Reader Service Card

CONTINUED ON PAGE 186

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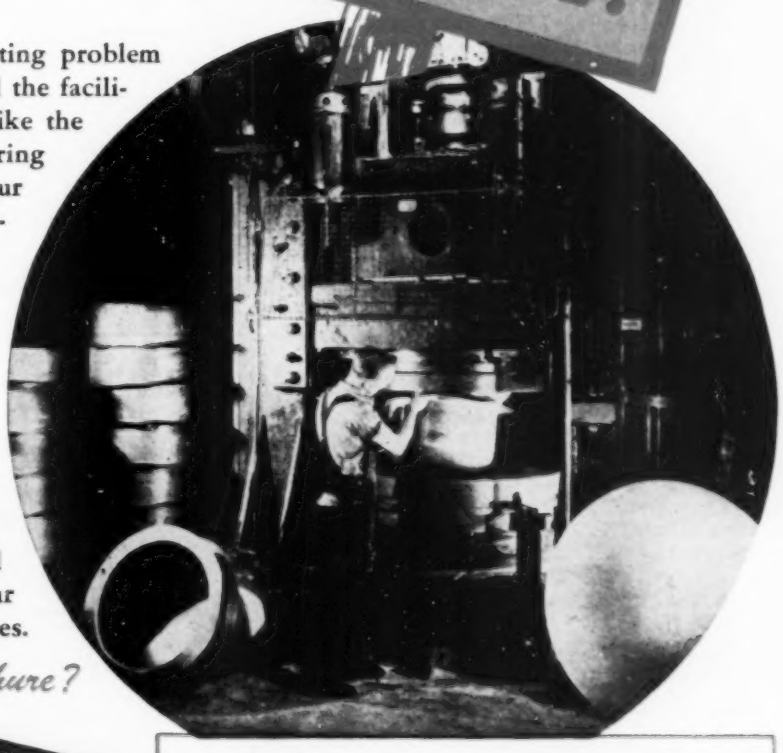
ENGINEERING SKILL and FACTORY FACILITIES FOR HIRE!



● Have you a metal-fabricating problem that's outside your field or beyond the facilities of your plant? Would you like the help of expert, experienced engineering and designing skill to develop your idea into a finished product, or improve an old product?

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- SPRAY FINISHING
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A complete machine shop and tool and die department.

GEUDER, PAESCHKE & FREY CO., 1505 W. St. Paul Ave., Milwaukee 3, Wis.

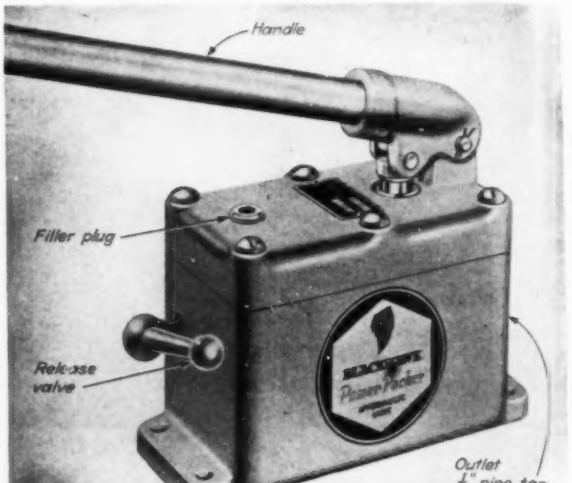
Hydraulic Pump Fits Close Quarters

Three models of this reservoir-pump have been designed with the same outside dimensions. Maximum pressures obtained are 6,000, 5,000 and 2,500 psi. The complete unit weighs 15 lb with handle. The housing itself measures $7\frac{7}{8}$ x $3\frac{1}{2}$ x $4\frac{5}{8}$ in. high.

All three models are similar. Pumps are single acting; needle cone release valves are used. Reservoir capacity can be increased from the 45 cu in. standard to 65 cu in. by the addition of a breather. Any mounting, vertical or horizontal is permissable, and the handle may be bent for operating convenience.

Standard design gives easier maintenance. At the outlet end of the housing a single plug gives access to all the ball check valves in the unit. The release valve handle adjusts to meet space or convenience requirements. Held to the pump by a positive lock, the handle can not slip out in operation. Handle is $17\frac{3}{4}$ in. long and has 45 deg of travel, from the horizontal upward. This assumes a straight handle and horizontal mounting. In an emergency, 10W oil can be substituted for the manufacturer's specified hydraulic fluid. The release handle can be adjusted to fit space requirements. The three new models represent an extension of the manufacturers standard line of larger capacity units. Maximum pressure is unchanged, but overall size has decreased.

Blackhawk Mfg. Co., Milwaukee 1, Wis.



Maximum Operating Pressure (psi)	6000	5000	2500
Displacement Per Stroke	0.15033	0.19635	0.37122
Diameter of Pump Piston, in.	7/16	1/2	11/16
Handle Effort at Maximum Pressure, lb.	57	61	58

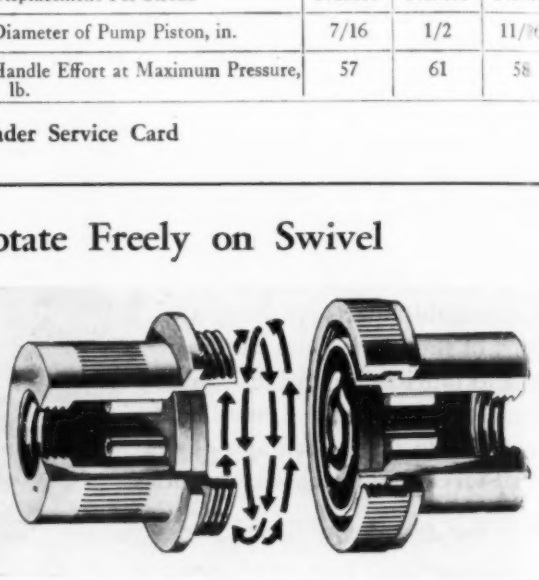
Circle No. 23 on Reader Service Card

Electrical Connections Rotate Freely on Swivel

Electrical connections can rotate freely in either direction without interruption of current or mechanical binding in this new unit. A screw housing encloses circular electrical contacts. Made with 2, 3 and 4 wire contacts, the joint is approved by the Underwriters Laboratory.

The 2 wire unit is $1\frac{15}{16}$ in. OD by $1\frac{3}{4}$ in. long. The 4 wire contact measures 2 in. OD by 2 in. long. Mounting holes carry $\frac{1}{8}$ to $\frac{1}{2}$ in. pipe or machine threads. Electrical rating is 660 w, 250 v. Bodies are made of brass, steel, aluminum, stainless steel, or any specified material.

Robboy Electric Mfg. Co., 2062 E. 70th St., Cleveland, Ohio



Circle No. 24 on Reader Service Card

NEON PILOT LIGHT mounts through $\frac{1}{2}$ in. diameter hole and extends $1\frac{1}{2}$ in. behind panel. Lamp is guaranteed for 10,000 hours at 75-150 v ac or dc. Nylon housing is special. Industrial Devices, Edgewater, N. J.

Circle No. 25 on Reader Service Card

CARBON BEARINGS are self-lubricating up to 800 F at relatively high speed. The material is also applicable to rings, seals and slides. Morganic Inc., 3304 48th Ave., Long Island City 1, N. Y.

Circle No. 26 on Reader Service Card

SPLASH-PROOF MOTORS are built with single or double reduction gears, with speeds from 20 to 780 rpm, in $\frac{1}{2}$ to 15 hp ratings for horizontal floor mounting. Sterling Electric Motors, Inc., Los Angeles 22, Calif.

Circle No. 27 on Reader Service Card

LOCK NUT holds hexagon insert on top of 1 piece cold forged body with six folded side projection petals. Furnished for No. 4 to $\frac{3}{8}$ in. screws. Townsend Co., New Brighton, Pa.

Circle No. 28 on Reader Service Card

CONTINUED ON PAGE 187

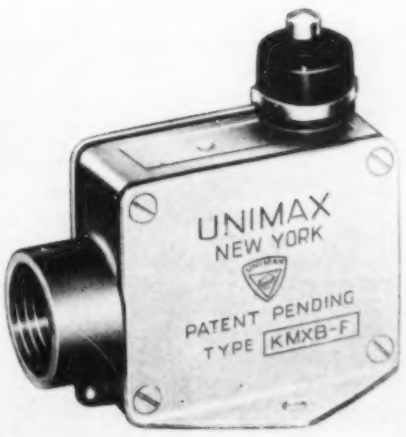
UNIVERSITY OF MICHIGAN LIBRARIES

Neoprene Seals Splash-Proof Precision Snap Switch

For protection against moisture, dust, and splashing oil and water, a neoprene "boot" and neoprene side gaskets effectively seal the metal case of the snap-acting precision switch. The "boot" is sealed to the actuating plunger and the entire assembly is integral with the metal case. The neoprene seal assembly is integral with the metal case. The seal retains its flexibility at temperatures from —65F to +160F.

The basic switch slips onto two positioning studs attached to one of the removable side plates. Three mounts are available: one with two side mounting studs; a second with two tapped bottom bosses for base mounting; a third for from-the-top base mounting, upright or inverted. Electrical ratings for the standard switch are 15 amps, 125 v a-c, 5 amps 250 v a-c, non inductive. Another model is designed for 15 amps 250 v a-c, 20 amps 125 v a-c, or ¾ hp 125, 250, 460 v a-c.

Unimax Switch Div., W L Maxson Corp., 461 W. 34th St., New York 1, N. Y.



Circle No. 29 on Reader Service Card

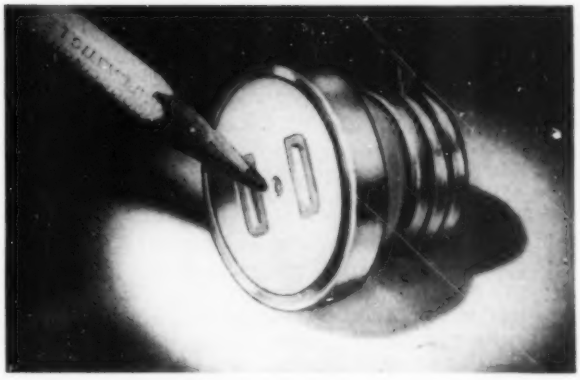
Nylon Plug Body Glows When Juice Is On

A neon glow lamp built into the body of the plug-in receptacle produces a light to facilitate connection and also serves as a pilot light to indicate the presence of power. The new unit is designed primarily for industrial testing applications where constant use causes wear and breakage. The pilot light immediately indicates power failure or a faulty plug.

The plug body is made of nylon and is designed for rough usage. Life of the neon lamp is estimated at 25,000 hours. Power consumption is less than $\frac{1}{25}$ watt.

In testing equipment through this plug, the operator can immediately distinguish between a faulty piece of equipment and a line power failure or a faulty plug. With the lamp on, any faults must be with the equipment under test.

Industrial Devices, Inc., Edgewater, N. J.



Circle No. 30 on Reader Service Card

TEST JACK gives front panel test point for any internal voltage. Measures ⅜ in. diameter by 1 in. long and passes fungus and salt spray tests. Lead wire connects to solder tab. Alden Products Co., Brockton 64, Mass.

Circle No. 31 on Reader Service Card

400 CYCLE MOTORS with ratings up to 2 hp are designed for aircraft applications. High efficiencies and power are featured, with high power to weight ratios. Integral a-c brakes provided. Lear Inc., Electro-Mechanical Div., Grand Rapids 2, Mich.

Circle No. 32 on Reader Service Card

LOW-TENSION CABLE withstands high heat, is claimed inert to all known solvents, and has greater fungus and abrasion resistance than braided cotton. Supplied in No. 8 to 20 gage sizes. Packard Electric Div., GMC, Warren, Ohio.

Circle No. 33 on Reader Service Card

INDUSTRIAL TIRES have been developed with improved wear and shock absorbent qualities. New contour and special rubber have been used. Monarch Rubber Co., 62 Lincoln Park, Hartville, Ohio.

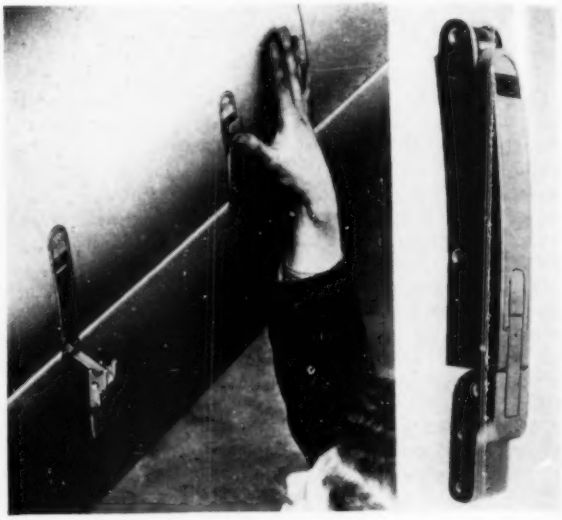
Circle No. 34 on Reader Service Card

PRIMARY BATTERY, non-renewable, air-depolarized, add-water type, is offered in 1,000 amp-hour model measuring 8¼ by 7¼ by 9½ in. Railroad switch lamp is lighted for 10 months. Thomas A. Edison, Inc., Primary Battery Div., Bloomfield, N. J.

Circle No. 35 on Reader Service Card

CONTINUED ON PAGE 188

Light Cowl Latch Take Heavy Loads



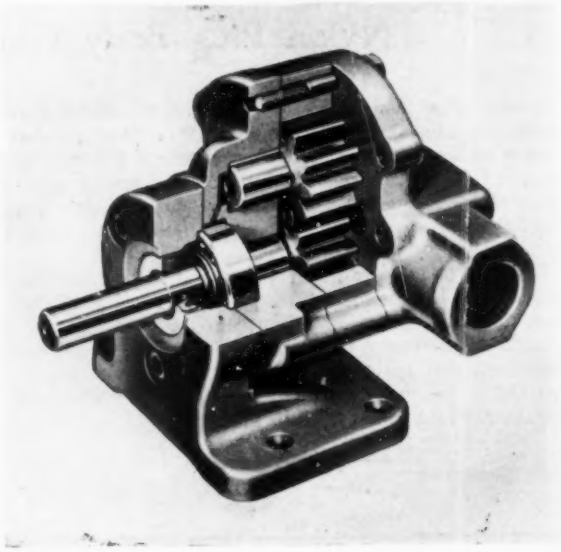
A smooth exterior on flat or curved surfaces is provided by this cowling latch. The device has been in service abroad for a number of years. The latch weighs from 6 to 8 oz and is designed to carry an ultimate load of approximately 7,000 lb in tension and 9,000 lb in shear. Four sizes accommodate all cowling curvatures from 23 in. radius to flat, and are suitable for either hinged or detachable panels. Maximum fitting error throughout this curvature range is held within 0.017 in. Total tension adjustment is $\frac{1}{2}$ in. and cowling pull-up is $\frac{3}{4}$ inches.

Latches are comprised of two units, a latch housing and a toggle assembly, and only four bolts are required for installation. There are no loose parts. When under load, the latch is self-closing over dead center. The cowling must be securely fastened before the toggle lever will assume its flush position, thus giving a check on latches.

Simmonds Aerocessories, Inc., Tarrytown, N. Y.

Circle No. 36 on Reader Service Card

Gear Pump Impellers Carry No Thrust Loads



A hydraulic pump line has been designed for 1,000 psi, with capacities of 1.5 to 46.8 gpm at 1,800 rpm. A special keying arrangement transfers torque without thrust loads from the drive shaft to gear impellers. Keys consist of balls placed in three axial grooves around the periphery of the shaft and the inside bore of the impellers. The gears float in the pump cavity regardless of lateral movements of the shaft.

Trapped fluid is utilized to pressure-lubricate shafts, bushings and bearings. This permits use of journal bearings. Special alloy bronze bushings capable of withstanding heavy loads at high temperatures, alloy iron housings and hardened steel gears and shafts are used. The shaft seal is designed for sealing with minimum horsepower loss. A ball bearing absorbs thrust and radial loads imposed by drive. Pumps are internally drained.

Adel Precision Products Corp., 10855 Van Owen St., Burbank, Calif.

Circle No. 37 on Reader Service Card

Electro-Magnet Engages Low Power Clutch

Designed for servos and other devices where rapid engagement and release is required, this compact magnetic clutch has a capacity of 0.20 hp with an input of 3 watts. When the electromagnet is energized, the cork faced driven disk is clamped between two driving members. For high torque transmission, the driven disk is faced on both sides. Splined bore in driven disk permits coupling to the load. Non-rotating electromagnet with flange mounting eliminates slip rings and auxiliary bearings. Driving plates are ventilated. Maximum torque is 4 lb. Maximum speed is 3,600 rpm. Input is 3 w d-c or rectified a-c.

Barber-Colman Co., Rockford, Ill.

Circle No. 38 on Reader Service Card

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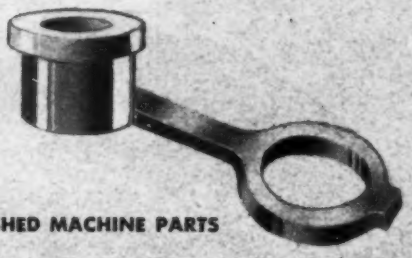
QUALITY



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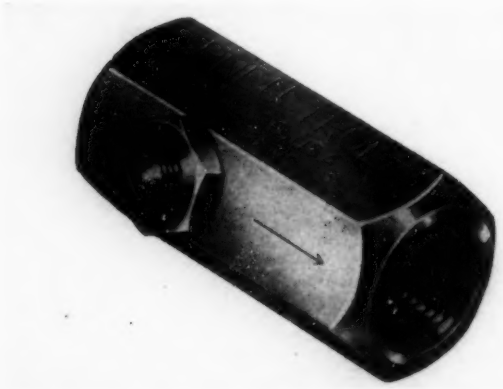
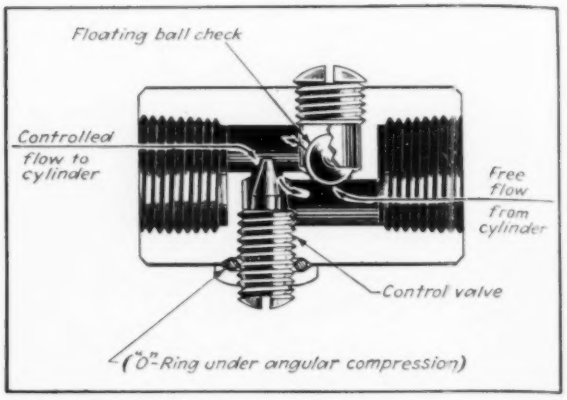
Flow Valve Adjusts Cylinder Speed on Power Stroke





A stainless steel ball check allows free flow of air or hydraulic fluid out of an actuating cylinder, while a variable needle orifice is built into the same control valve to adjust actuating speed by metering the input of fluid.

Plain check valves or fixed orifice control valves are also offered. Up to 1,000 psi, bodies are machined from brass bar stock. From 1,000 to 5,000 psi, steel is used. All internal parts are stainless steel. Pipe sizes $\frac{1}{8}$, $\frac{1}{4}$, $\frac{3}{8}$, $\frac{1}{2}$ and $\frac{3}{4}$ in. are available.

Pneu-Trol Devices, Inc., 3122 N. California Ave., Chicago, Ill.

Circle No. 39 on Reader Service Card

Electronic Timer Switch Controls Automatic Machines

Process control and machine timing offer applications for a new electronic timer switch. A number of points in the internal circuit are brought out through front panel terminals in order to provide flexibility in application. Provisions are made for four basic types of timing: interval, delayed action, automatic repeat and programming. Variations on the fundamental cycles are possible.

One tube and one relay are employed. The circuit is self-compensating and voltage change or tube replacement do not effect the original 2 percent accuracy. The assembly weighs 8 lb and current is 115 or 230 v, 50/60 cycles. Ambient temperature and humidity are given.

Photoswitch, Inc., 77 Broadway, Cambridge 42, Mass.

SPECIFICATIONS

Supply	115 or 230 v, 50/60 cycles
Supply Power	25 watts
Output	Two single-pole, double-throw switches
Relay Contact Rating	10 amps, 115 v a-c Non-Inductive
(SPDT)	5 amps, 230 v a-c Non-Inductive
Auxiliary contracts	1 amp, 115 v a-c Non-Inductive
	.5 amp, 115 v. a-c Non-Inductive
Ambient Temperature	32–150 F
Ambient Humidity	20–95 percent
Net Weight	8 lb.

Circle No. 40 on Reader Service Card

DRAWING SCALE offered in triangular form with scales to make enlarged drawings in ratios of 1-1, 2-1, 3-1, 4-1, 5-1, 8-1 and multiples of ten. Lindly and Co., 80 Herricks Rd., Mineola, N. Y.

Circle No. 41 on Reader Service Card

SMALL MOTOR offered for 400 cycles, $\frac{1}{30}$ hp, 115 v, 3 phase, built to Air Force Spec. 32590. Motor weighs 9 oz and is 2 in. in diameter by $1\frac{5}{8}$ in. long. Efficiency of 70 percent claimed. Mission Electric Mfg. Co., 132 W. Colorado St., Pasadena, Calif.

Circle No. 42 on Reader Service Card

COPPER OXIDE RECTIFIER with reverse resistance of 1 megohm per plate is intended for circuits in which low leakage and maximum stability are essential. Measures $\frac{1}{4}$ in. in diameter by $\frac{5}{8}$ in. long. Rated to 5 milliamp d-c. Bradley Laboratories Inc., New Haven 10, Conn.

Circle No. 43 on Reader Service Card

CIRCULAR HEATING UNITS are offered in narrow bands to supply 40 w per sq in. Breakdowns are localized. Applications in plastic molding machines. Watlow Electric Mfg. Co., 1320 N. 23rd St., St. Louis 6, Mo.

Circle No. 44 on Reader Service Card

INSTRUMENT TUBING is available in $\frac{1}{4}$ and $\frac{3}{4}$ in. diameters with $\frac{1}{32}$ in. thick thermoplastic coating, immune to moisture, salts, oils, acids and alkalies. Samuel Moore & Co., Mantua, Ohio.

Circle No. 45 on Reader Service Card

CONTINUED ON PAGE 192

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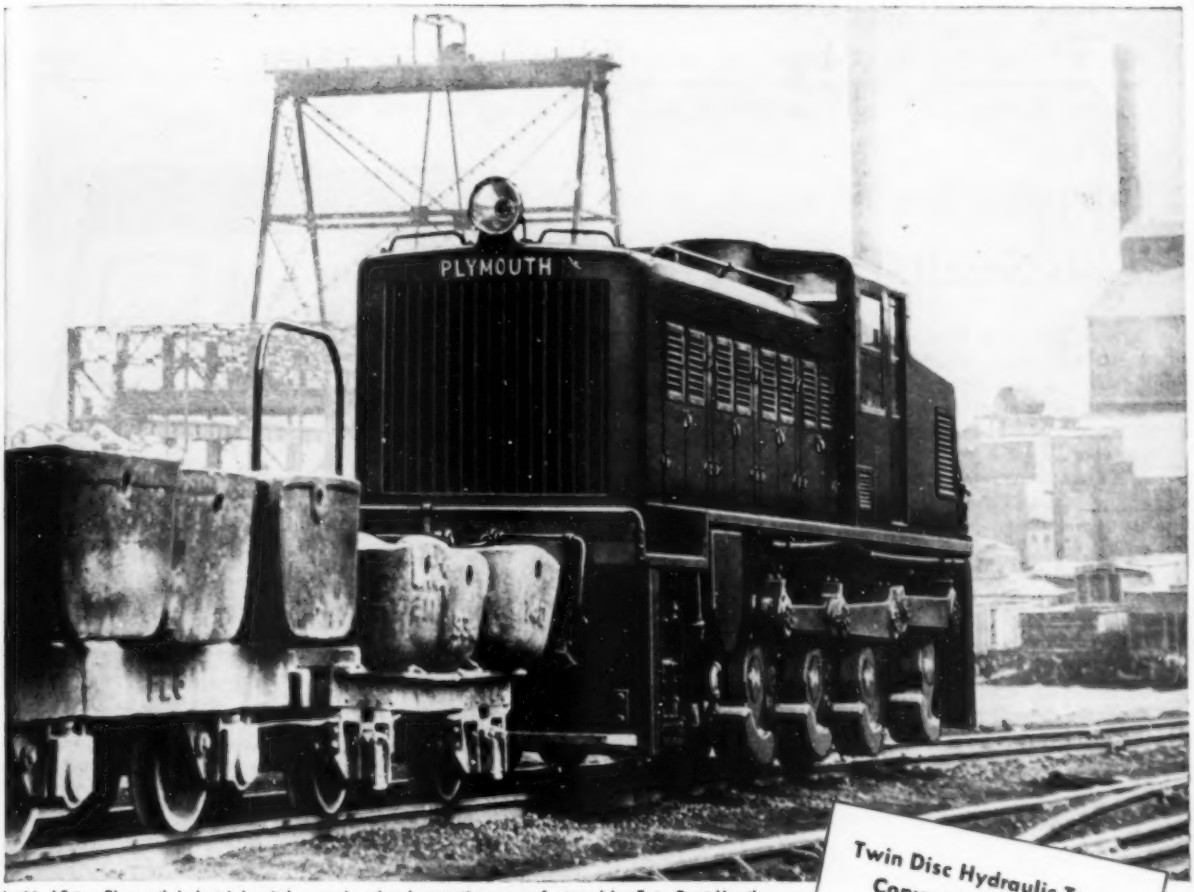


In this 65-ton Plymouth industrial switcher, as in other locomotives manufactured by Fate-Root-Heath, in the 12 to 70-ton class, you'll find Twin Disc Hydraulic Torque Converters . . . fluid power to eliminate rough starts, to handle more cars per day, and handle them cheaper.

Twin Disc Hydraulic Torque Converter Advantages

1. All shocks and vibrations absorbed. 2. Prime mover can't be lugged down or stalled. 3. Engine torque multiplied five times at stall. 4. Infinitely variable reduction ratio automatically adjusts to load conditions.

No Rough Starts Here

S-m-o-o-t-h-l-y . . . s-m-o-o-t-h-l-y . . . that's the way powered equipment picks up under full load when the engine delivers its power through a Twin Disc Hydraulic Torque Converter. There's no gear-shift guesswork: the Torque Converter offers an infinitely variable number of gear ratios *automatically.* Under starting or stall conditions, it *multiplies* engine torque as much as five times to increase tractive, hoisting, or other effort.

On this 65-ton Flexomotive Plymouth industrial locomotive a 16,000 series Twin Disc Torque Converter was specified to assure smooth, uninterrupted acceleration—output torque automatically adjusted to load requirements—eliminating the use of heavier, more expensive diesel-electric units. A planetary transmission with high, low and reverse is coupled with the Torque Converter to comprise the entire transmission.

This smoothness, this automatic selection of the right gear ratios, means more work done, longer equipment life. That's why more and more equipment manufacturers are switching to Twin Disc Hydraulic Torque Converters (Lysholm-Smith type)—available in 40 to 1,000 hp capacities. Write today for Bulletin No. 135-C.

TWIN DISC
CLUTCHES AND HYDRAULIC DRIVES

Clutches & Hydraulic Drives

TWIN DISC CLUTCH COMPANY, Racine, Wisconsin • HYDRAULIC DIVISION, Rockford, Illinois

BRANCHES: CLEVELAND • DALLAS • DETROIT • LOS ANGELES • NEWARK • NEW ORLEANS • SEATTLE • TULSA

Portable Instrument Tests Small Tubes

A new laboratory type instrument is built to test all sub-miniature tubes, including television. The scale reads directly in microhms. A separate meter is provided to permit adjustment of line voltage while a tube is under test and a plate milliammeter can be inserted to read plate current.

The instrument has two transformers to test future life of the tube and three a-c signal voltages, 0.25, 0.5 and 2.5v are available in addition to the d-c grid bias and plate and screen voltages. Provision is made for self-bias and for vernier adjustment of bias. This feature is accomplished by a 200 ohm rheostat with calibrated dial, by-passed by a 1,000 mfd capacitor whch can be inserted in the cathode circuit by operating a switch. The leatherette carrying case measures 17 by 18 by 18½ in., and the entire instrument weighs 31 pounds. Operation is on 110-130v a-c lines. With the outside line current adjusted, the instrument gives the important characteristics of the tube under test simultaneously on one set of dials.

Hickok Electrical Instrument Co.,
10544 *Dupont Ave., Cleveland* 8, *Ohio*

INSTRUMENT is designed to test all sub-miniature tubes. Voltages of 0.25, 0.5 and 2.5 are used with d-c grid bias and d-c plate and screen voltages. A separate meter reads line voltage for accurate adjustment. Plate current of the tube under test is also shown. The main tube testing scale reads directly in microhms. Leatherette carrying case weighs 31 pounds.

Circle No. 46 on Reader Service Card

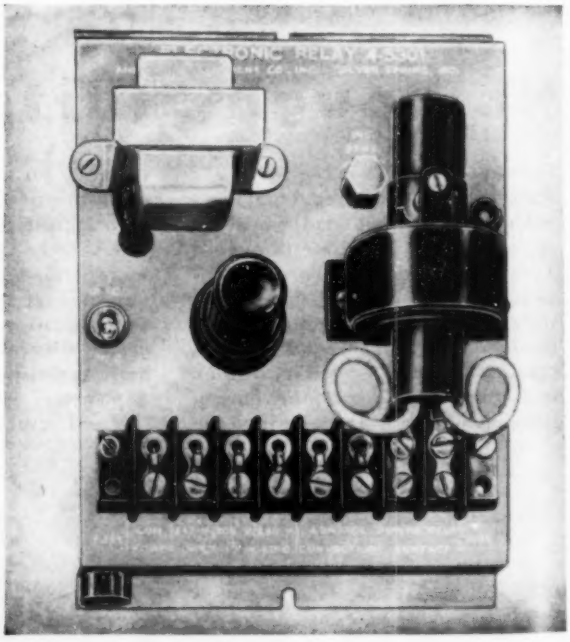
Electronic Relay Has High Power Gain

A new electronic relay has been designed for a range of operation covering motors, alarms, instruments, thermal sensitive resistors, and level and motion control.

Power gain is 200 million and the relay can be operated at 117,208 or 230 v, 60 cycle. Continuous adjustment is provided from 3 microamps to 1 milliamp control current with operating resistances of 1700 ohms to 1.5 megohms. A built-in switch converts the relay from normally open to normally closed. The relay is uneffected by ±10 percent line voltage variation and handles loads of 20 amps at 250 volts. Control leads up to 2000 ft long may be used. The power relay is the mercury plunger type and the enclosure is fitted with BX cable connectors and insulated bushings.

The circuit permits operation of the thyratron tube below half its rating and at reduced filament voltage. Applications are possible on instruments where contact current, wear and deterioration are important factors.

American Instrument Co., Inc., Silver Spring, Maryland



Circle No. 47 on Reader Service Card

CONTINUED ON PAGE 194

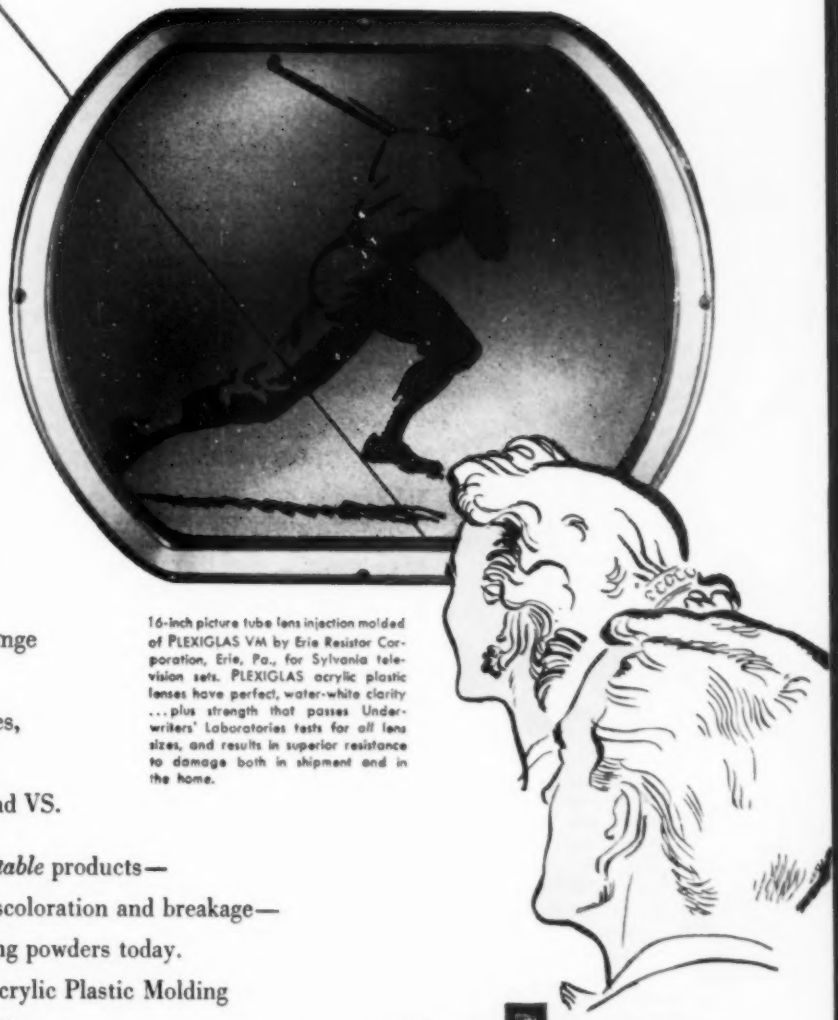
UNIVERSITY OF MICHIGAN LIBRARIES

PLEXIGLAS

means "Good Looking" because it's crystal clear

When you want customers to look *at*—or look *through*—your product, specify PLEXIGLAS molding powder. This acrylic material makes clear or colored decorative pieces gleam like jewels. And in lenses, transparent sections, housings, true optical clarity is assured.

The excellent flow characteristics of PLEXIGLAS powders permit satisfactory molding over a wide range of temperatures and pressures. We offer 3 different injection grades, each developed to meet specific requirements—PLEXIGLAS V, VM and VS.



16-inch picture tube lens injection molded of PLEXIGLAS VM by Erie Resistor Corporation, Erie, Pa., for Sylvania television sets. PLEXIGLAS acrylic plastic lenses have perfect, water-white clarity ... plus strength that passes Underwriters' Laboratories tests for all lens sizes, and results in superior resistance to damage both in shipment and in the home.

For injection molding of *durable, stable* products—resistant to high heat, weather, discoloration and breakage—get full details of PLEXIGLAS molding powders today. Our newest booklet, "PLEXIGLAS Acrylic Plastic Molding Powders", gives complete information.

SEND FOR YOUR FREE COPY TODAY

PLEXIGLAS *is a trade-mark, Reg. U. S. Pat. Off. and in principal foreign countries.*
Canadian Distributor: Crystal Glass & Plastics, Ltd.
54 Duke St., Toronto, Ont.

CHEMICALS FOR INDUSTRY

ROHM & HAAS COMPANY

WASHINGTON SQUARE, PHILADELPHIA 5, PA.

Representatives in principal foreign countries

For HEAVY DUTY SERVICE

Drive Shafts of

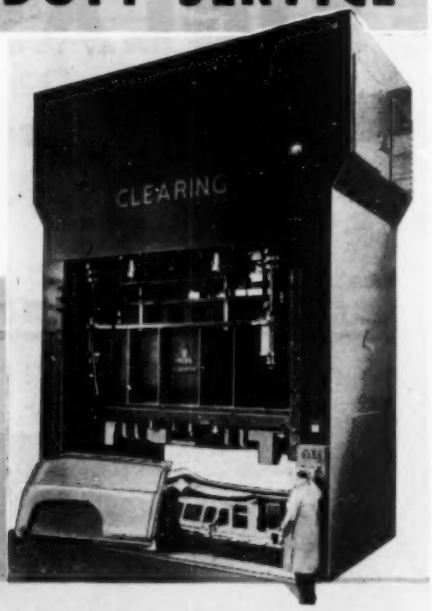
CLEARING PRESSES

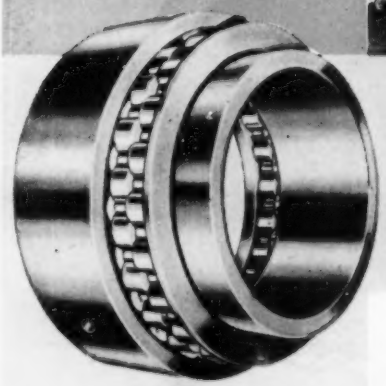
rotate on

ORANGE

"STAGGERED"

ROLLER BEARINGS





Clearing Hydraulic and Mechanical Presses exert accurate pressures up to thousands of tons—turn out huge sections of car bodies in a single stamping. Clearing engineers find Orange Staggered Roller Bearings best suited to withstand the severe service in the drive shafts of Clearing Presses and have been using these bearings with excellent success for 10 years.

How unique staggered and meshed roller design provides greater load carrying capacity

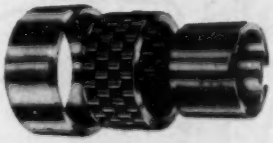


Illustration above shows how load is divided over many short rolls instead of a few full length rolls. At left, end views of Staggered bearing and conventional bearing show how staggered roller design brings a multiplicity of contact points within the loaded zone.

DO THE WORK OF LARGER-SIZE CONVENTIONAL BEARINGS

Wherever designs call for bearings combining extreme ruggedness, high load carrying capacity and precision running—be sure to study the capacities of Orange Staggered Roller Bearings. Size for size, they carry higher loads than conventional roller bearings. Used in new designs, you can carry the required load with smaller size Staggered bearings and save space. Used as changeovers in present equipment, you gain extra load capacity and longer service life.

Orange Staggered Roller Bearings are available in a full range of interchangeable sizes—in the 200 and 300 series, from 305 to SW-244, and in the 5200 and 5300 series, from 5306 to 5334. Also with notched-type inner race, supplied with separable outer and inner races, as required. Special sizes to order.

WRITE FOR ENGINEERING DATA FOLDER

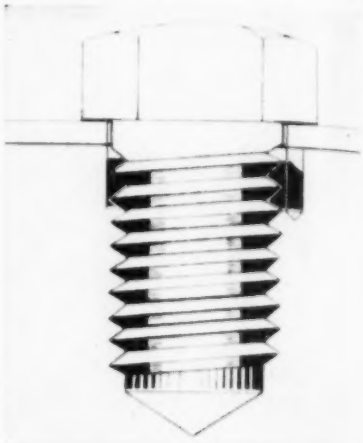
showing construction, types, sizes, capacities, installation data, etc.

ORANGE ROLLER BEARING CO., INC.

551 Main Street, Orange, N. J.

Elliptical Ring Locks Stud

The stud forces an elliptical spring steel retainer in a round hole, giving locking pressure on the threads of the unit. The retainer floats in the counter-



bore, bearing no load. The entire stress is carried by the threads in the tapped hole. Studs are available in NC and NF threads in standard sizes from $\frac{3}{8}$ to 2 inches.

Security Locknut Corp., 1815 N. Long Ave., Chicago 39, Ill.

Circle No. 48 on Reader Service Card

Air Accumulator Stores Fluid At High Pressure

Low-pressure-air-operated accumulators have been designed to eliminate the danger of handling high pressure gasses and store fluids at up to 10,000 psi for use when required.

Ordinary plant air at 60 to 200 psi is used as the compressible medium. In pump circuits, oil flows from the pump into the accumulator during the non-demand pump cycle. The oil, under the high pump pressure, works against the low-pressure compressible (air) medium. The oil is thus "stored" at pressures up to 10,000 psi, for either immediate or later discharge, usually at high flow rates of short duration. They are generally used in this capacity on die casting machines, plastic molding presses, heavy machine tools, etc.

The all metal construction of these accumulators is claimed to adapt them for applications in which severe shock loads are encountered, particularly when the accumulator must completely unload itself in stopping and reversing the motion of a moving mass. These units can be unloaded and reloaded at will and can be advantageously employed in non-pumping applications, such as the loading of hy-

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Slower Speeds to fit Your Needs

Wagner Gear Motors

Now available in ratings down to 1/20 horsepower

Where "slower than motor" speeds are required — on agitators, conveyors, scrapers, or in a host of other applications—you can obtain *positive reduction with the right power* by using Wagner Gear Motors.

These gear motors combine famous Wagner motor dependability with nationally-known gear units to give you compact, efficient gear motors that can help reduce costs, improve your product, and increase your plant operating efficiency by eliminating the need for belts, pulleys or chains.

The expanded Wagner line of gear motors now includes ratings from 1/20 to 50 horsepower and with gear ratios to meet every requirement. Their flexibility of mounting provides an exceptionally large number of combinations to meet particular needs.

Twenty-nine branches, located in principal cities, and manned by trained field engineers, are at your service. Write for information on the complete line of Wagner Gear Motors.

Wagner Electric Corporation
EST. 1891

WAGNER ELECTRIC CORPORATION
6406 Plymouth Ave., St. Louis 14, Mo., U.S.A.

ELECTRIC MOTORS · TRANSFORMERS · INDUSTRIAL BRAKES
AUTOMOTIVE BRAKE SYSTEMS — AIR AND HYDRAULIC

BRANCHES IN 29 PRINCIPAL CITIES

The doctor held his breath. An iron lung had stopped. For four years it had kept a little girl alive. Now . . . ? Feverishly, hospital attendants readied a temporary lung. But it would be only a matter of hours before substitute lung and young life would snuff out together.

Five hundred miles away in the Winsmith plant, the phone jangled with an urgent question. Could they? They'd *have* to! The clock on the wall said three. Experts cooly raced the hands of the clock and at four, the gravely needed speed reducer had been assembled and flown on its way.

That night, the little patient . . . *everyone* breathed easier. Winsmith had won a race against time.

Unusual? Not altogether, especially when an equipment manufacturer has the foresight to install a *standard* reducer. Yes, meeting and beating tough challenges is an old Winsmith habit, thanks to standardization and management's responsiveness to customers' problems.

Your speed reducer problems may not be a matter of life or death, but they're often a matter of profit or loss. Winsmith can help you solve these problems to your best interests . . . time-wise and cost-wise. One of the reasons . . . industry's widest range of standardized types and sizes up to 85 H.P. Another . . . the large inventory of standard parts ready for quick assembly. Together, they enable the Winsmith field engineer in your territory to prescribe to your needs right out of stock from one of these 3 basic types:

FREE CATALOG
Handbook No. 148 with complete engineering data. Write.

WHS WINSMITH SPEED REDUCERS

AGMA
WINFIELD H. SMITH CORPORATION

draulic die cushions, and in other applications where unloading for initial filling or for feeding new stocks is necessary, such as in providing the holding pressures for cylinders used on strip mill rolls in steel processing, on tanning squeeze or wringer rolls in leather processing, or on pressure rolls of textile machinery.

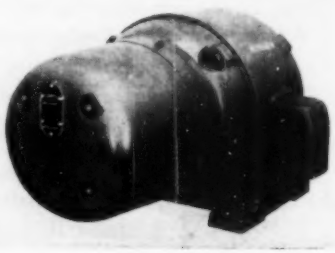
By changing their circuit connections, these accumulator units can be employed as "boosters", with either plant oil or air line pressure used as the input medium in developing up to 10,000 psi hydraulic output pressure for operation of hydraulic work cylinders. The units are offered in a variety of convenient sizes and mounting styles, and are built up from standard air and hydraulic cylinder components that permit fast production of any model.

Miller Motor Company, 4027 North Kedzie Ave., Chicago 18, Illinois

Circle No. 49 on Reader Service Card

Brake Motors Hold On Power Failure

Motors up to 20 hp in the manufacturers standard line are now available with an explosion proof, electrically operated brake as unit apparatus. Brake combinations have been selected to operate at 100 and 150 percent of full load motor torque. An adjustment nut sets the torque for specific



load conditions, allowing operation below maximum rated valve. A wear indicator is visible through a plastic window, which may be removed for access to the combination hand release and wear indicator. A screwdriver adjustment compensates for wear.

General Electric Co., Schenectady 5, N. Y.

Circle No. 50 on Reader Service Card

Flat Resistor

A new flat, metal-encased, wirewound power resistor is specifically designed for business machines and other exacting equipment requiring depend-

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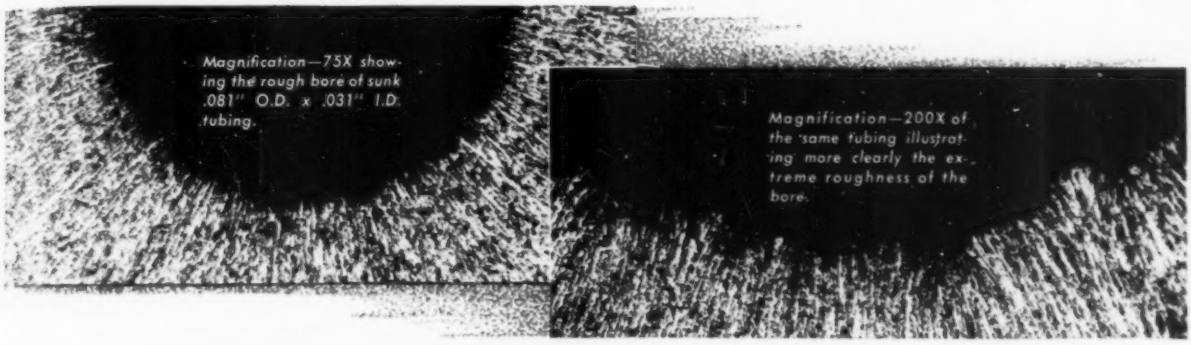
WHICH TUBE WOULD YOU CHOOSE?

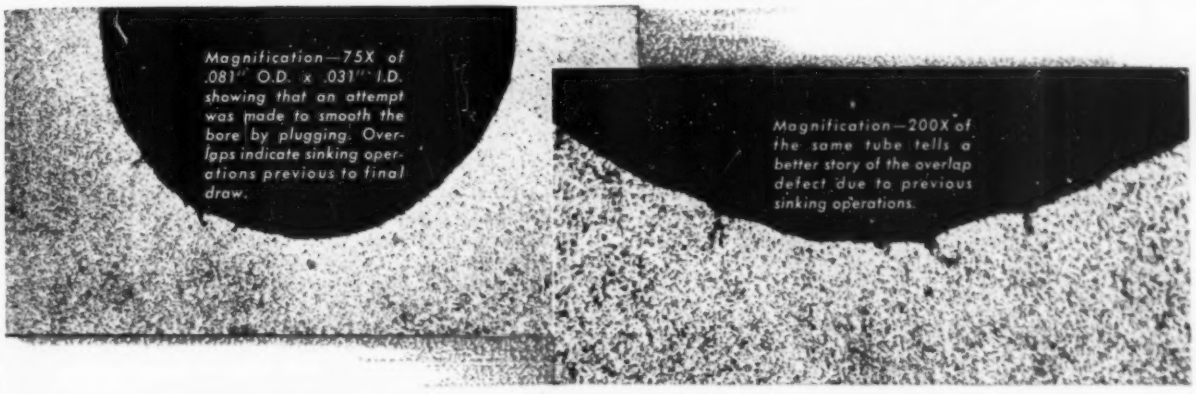
These photomicrographs show the transverse sections of three brands of capillary tubing.

They tell their own stories and prove the truth of our statement that "There IS a difference in tubing."



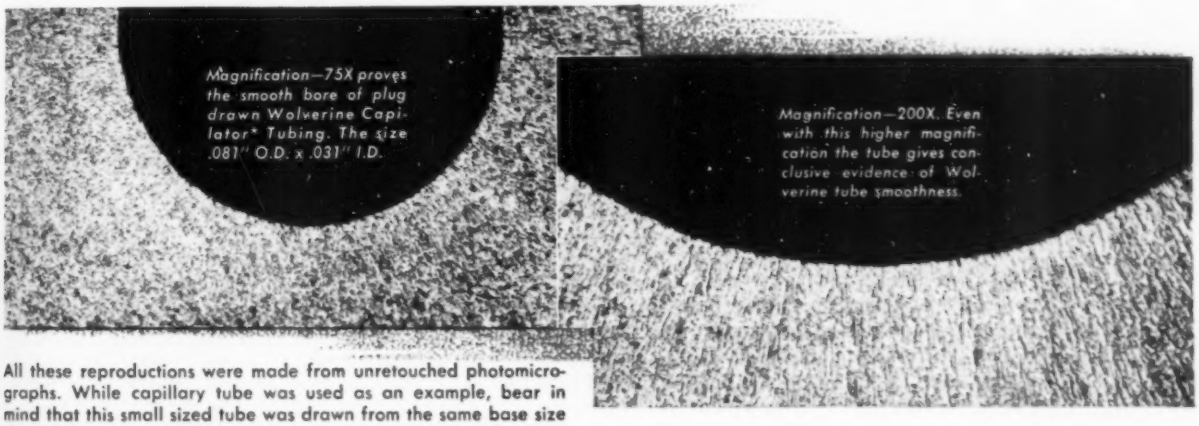
Magnification—75X showing the rough bore of sunk .081" O.D. x .031" I.D. tubing.

Magnification—200X of the same tubing illustrating more clearly the extreme roughness of the bore.



Magnification—75X of .081" O.D. x .031" I.D. showing that an attempt was made to smooth the bore by plugging. Overlaps indicate sinking operations previous to final draw.

Magnification—200X of the same tube tells a better story of the overlap defect due to previous sinking operations.



Magnification—75X proves the smooth bore of plug drawn Wolverine Capilator Tubing. The size .081" O.D. x .031" I.D.*

Magnification—200X. Even with this higher magnification the tube gives conclusive evidence of Wolverine tube smoothness.

All these reproductions were made from unretouched photomicrographs. While capillary tube was used as an example, bear in mind that this small sized tube was drawn from the same base size (3" O.D.) used to produce all other sizes of Wolverine tube. Thus this evidence of quality applies to ALL Wolverine tube, regardless of size.

Which tube would you choose? Of course, you'd choose the tube which would allow the smoothest flow of liquid or gas—and, in the long run, give you the most economical service. You'd choose dependable Wolverine tube.

*Reg. U.S. Patent Office

Let us send you literature telling about the use of tubing in your own applications.

WOLVERINE TUBE DIVISION

Calumet & Hecla Consolidated Copper Company
INCORPORATED

MANUFACTURERS OF SEAMLESS, NON-FERROUS TUBING

1415 CENTRAL AVE. • DETROIT 9, MICHIGAN

PLANTS IN DETROIT AND DECATUR, ALA.
Sales Offices in Principal Cities

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How to make PLANNED assembly savings PAY OFF!

STRONGER ASSEMBLY, TOO, IS CHILD'S PLAY

When the Bender Playground Co., maker of the Bender Stainless Steel Slide, was searching for a way to strengthen its product, P-K Self-tapping Screws were tried out. Results were so satisfactory that P-K Screws are specified today for every Bender Slide. Each slide contains 40 Type "A" Phillips Head Self-tapping Screws—air driven through steel into wood. "They hold better . . . don't come loose," reports the manufacturer. Why not invite a P-K engineer in to discuss your problem? You may be able to make similar savings.

Being on the lookout for better ways of doing old familiar jobs led this manufacturer to try Self-tapping Screws. Being, as well, a smart businessman—he naturally knew apparent savings do not always prove out. So, to make sure, he *wisely* chose P-K Self-tapping Screws. For performance, not price, is the only safe yardstick to use in figuring the ultimate cost of fastening methods.

Screws that don't Measure up cause costly Slowdowns

Screws that are the wrong type, that break or otherwise don't measure up, can quickly ruin your carefully *planned* savings. That's why successful manufacturers everywhere insist on Parker-Kalon Self-tapping Screws. Parker-Kalon originated the Self-tapping Screw. Throughout 35 years of manufacture, P-K has maintained its high quality standards—standards that keep assembly lines trouble-free.

P-K* has the RIGHT fastener for every job

Parker-Kalon makes a complete line of Self-tapping Screws. P-K is able to fit the fastener to the individual application—never requires you to fit the job to a certain type fastener. Find out for yourself why so many nationally known manufacturers say—"If it's P-K . . . it's O.K." Parker-Kalon Corporation, 200 Varick Street, New York 14, N. Y. Sold everywhere through accredited Distributors.

*TRADE MARKS REG. U.S. PAT. OFF.

TYPE A WITH PHILLIPS RECESSED HEAD
(Also made with slotted head)

P-K TRADE MARK

The Original
PARKER-KALON* SELF-TAPPING SCREWS

A TYPE AND SIZE FOR EVERY METAL AND PLASTIC ASSEMBLY

ability and minimum size. The resistor is space wound, has mica rather than phenolic insulation, and is encased in aluminum for mounting flat against a metal chassis to assure heat dissipation. At 175 C continuous operating temperature these units are rated for 7½ w in still air and 15 w when mounted on a metal chassis. The resistor is claimed to have safely dissipated as much as 35 w over long periods when mounted flat against a heavy metal plate, although such extreme loading is, of course, not recommended for normal use.

Shallcross Mfg. Co., Collingdale, Pa.

Circle No. 51 on Reader Service Card

Rubber Sandwich Cuts Vibration

Stud-type vibration isolators control transmitted vibrations in compression or shear. A rubber cylinder is bonded to steel end plates to which studs are welded. One stud is bolted to a foundation, the other to a piece of equipment. The rubber is bonded to the metal with breaking strength of ten times the working load, so that it is not necessary to anchor the studs within the rubber. This affords maximum dampening efficiency.

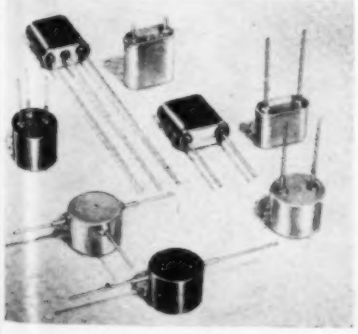
Available in 44 different sizes, for load ranges from 1# to 270# per mount, the isolators lend themselves to use with small motors, engines, lathes, air conditioning and lighting equipment.

Finn and Co., 2850 Eighth Ave., New York 30, N. Y.

Circle No. 52 on Reader Service Card

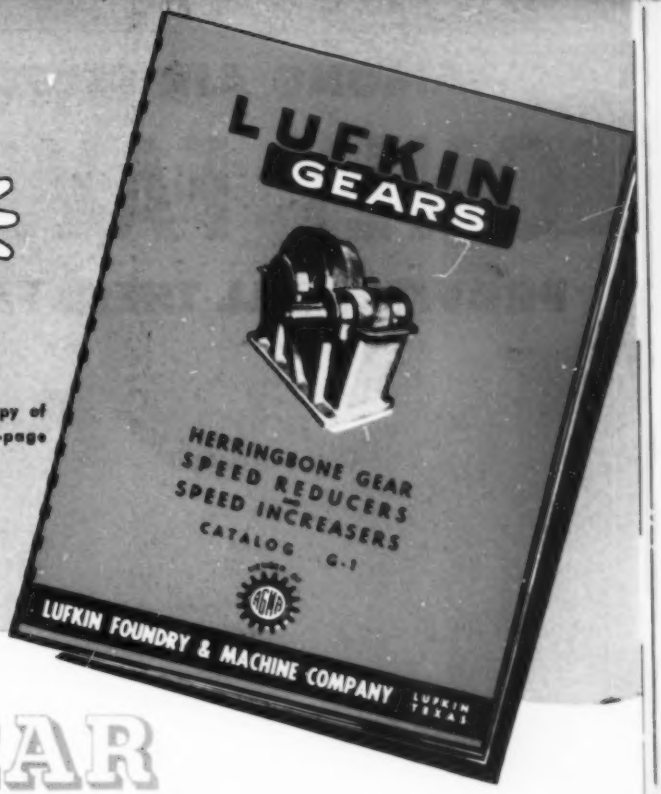
Bi-Metal Thermostats Sealed In Metal Enclosures

Corosion resistant metal cases are hermetically sealed around these new bi-metal disk and strip thermostats. Terminals are soldered into steel tubes



Lufkin

You can have a copy of this valuable 48-page catalog — SEE COUPON



GEAR

Reducer and Increaser Information

That Will Help You in Planning

FOR THE MOST . . . in Cost Saving

Dependable Product Performance

This Catalog, with the tremendously helpful information contained in its 48 pages, can be of great value to you. Numerous pages of photographs show LUFKIN Gears in the service of various industries, such as the oil industry—one of the toughest for gears. Photos of the various type Reducers and Increasers are shown with complete specifications. Pages are devoted to a "guide for selecting character of load" and a "service factors table." There are a number of pages on "selection of gear reducers." Horsepower rating tables are given as well as pages on load capacities and pages showing general dimensions. All this with other pertinent gear details make this catalog a dependable aid in your planning.

LUFKIN FOUNDRY & MACHINE CO., LUFKIN, TEXAS

Please send copy of your Catalog G-1.

NAME_____ TITLE_____

COMPANY_____

STREET_____

CITY_____

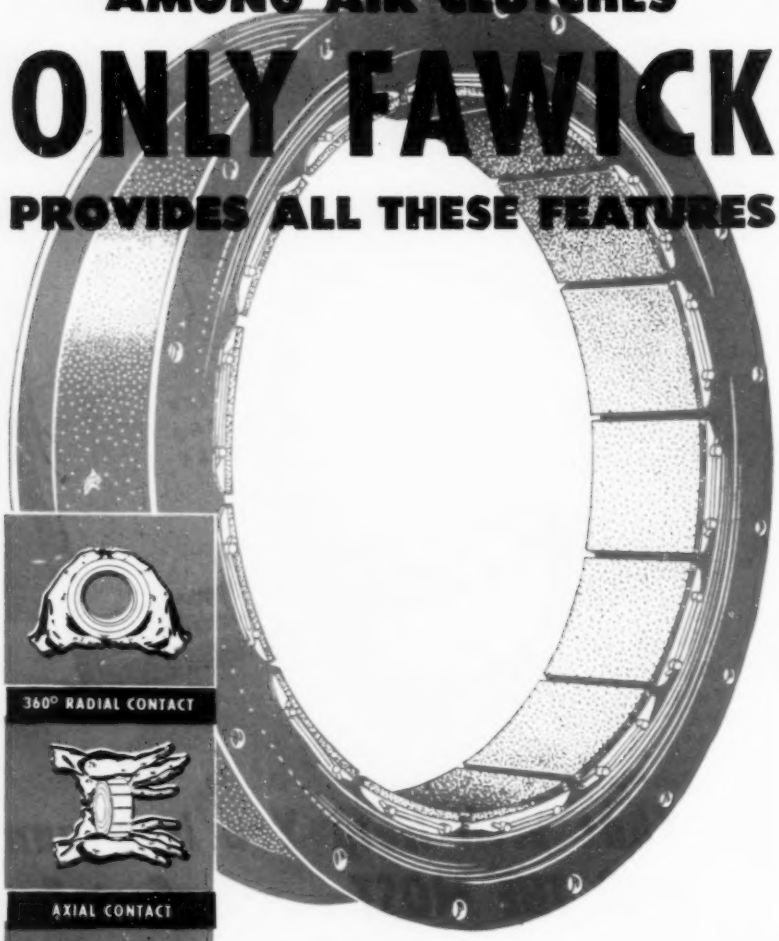
STATE_____

LUFKIN GEARS

AMONG AIR CLUTCHES

ONLY FAWICK

PROVIDES ALL THESE FEATURES



360° RADIAL CONTACT

AXIAL CONTACT

SHOCK ABSORPTION

CONTROLLED TORQUE

360° RADIAL CONTACT—Application of the operating pressure at the maximum diameter results in the greatest operating torque.

AXIAL CONTACT—Fawick design provides uniform-pressure, constant-velocity contact between the full width of the friction shoes and the drum.

SHOCK ABSORPTION—Operating torque is transmitted through the flexible rubber tube, dampening shaft vibrations and cushioning peak load shocks.

CONTROLLED TORQUE—Variation of operating pressure makes sensitive control of applied torque possible. Peak overload protection is assured at a selected air pressure.

For specific information on all advantages of Fawick Industrial Clutch and Brake Units, write to the Main Office, Cleveland, Ohio, for Bulletin 300.

All desirable clutch characteristics are built into Fawick Airflex units

FAWICK Airflex CO., INC.

9919 CLINTON ROAD CLEVELAND 11, OHIO

FAWICK CLUTCHES = = PEAK EFFICIENCY

having the same coefficient of expansion as the hard glass insulator bead surrounding them.

The new units are designed for service from —60 to 600 F in any precision device subject to dust moisture or corrosion. An electrically independent bi-metal thermal control element is claimed to eliminate artificial cycling. Special order units can be sealed in helium.

Stevens Mfg. Co., Inc., 69 So. Walnut St., Mansfield, Ohio

Circle No. 53 on Reader Service Card

Combination Gear Motor

A compact industrial motor drive combines a 1 to 10 ratio variable speed control with a gear head for slow speeds and high torques. It is available in $\frac{1}{4}$, 1/3, $\frac{1}{2}$ and $\frac{3}{4}$ hp; speed ranges are from 10 to 10,000 rpm. The frame is cast aluminum. A dial control permits speed changes by turning the handle less than one revolution for any variation. Slippage is prevented by a belt tensioner which automatically compensates for load. The gears are specially designed and have a hardness of 45/50 Rockwell C. Inspection of the interior is made by loosening four screws and removing the cover plate.

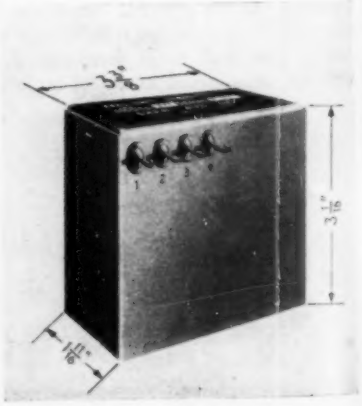
U. S. Electrical Motors Inc., 200 E. Slauson Ave., Los Angeles 54, Calif.

Circle No. 54 on Reader Service Card

Tuning Fork Gives Standard Frequency

A tuning fork assembly provides constant frequency sine wave voltage from a 28 v d-c source in this new light-weight frequency generator.

The new model consists of a miniature compensated tuning fork of high Q design, electronic driving means, suitable output transformer, thermo-



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New Townsend Tufflok Nut Is Tougher Three Ways To Give You Tighter Grip, Greater Safety, With Economy

You get a tighter grip, greater safety and economy with the new Townsend Tufflok nut. It gives you more service at less cost. The exclusive Tufflok design*, plus the materials in the insert and nut body combine to give you the toughest lock nut yet produced for any use where nuts must stay put under a variety of adverse conditions.

Here are the New Exclusive Features of the Townsend Tufflok Nut:

1. *Tough hexagon insert*—won't turn in its seat, gives you positive grip on the bolt at all times. Specially treated insert resists moisture, oil and drying out. Special inserts where special applications are required.
2. *Tough, one piece cold forged nut* has exclusive petal design to assure positive insert staking.
3. Tough threads *cut in cold forged steel* are stronger, spin on easier, resist galling, speed up application and also afford more reuse.

The quality, efficiency, and safety with economy, you get in the new Townsend Tufflok nut is made possible by the skill and experience gained by The Townsend Company as the oldest manufacturer of small cold formed fasteners in the United States. This know-how combined with the background of The Nylok Corporation in producing millions of lock nuts gives you the advantage of a reliable source for your lock nut requirements.

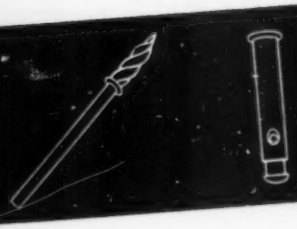
The new Townsend Tufflok nut with exclusive hexagonal treated insert and special cold forged body is available in machine screw sizes No. 4 through 3/8", cadmium or zinc plated. This line will be increased in the near future.

For longer service with greater safety and economy specify Townsend Tufflok Nuts. Write today for samples and complete information.

THE TOUGHEST LOCK NUT EVER PRODUCED

*Licensed under The Nylok Corporation patents. Tufflok is a trademark of the Townsend Company.

Call or write today to:



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COMPANY — ESTABLISHED 1816

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YOU SAVE

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your specifications

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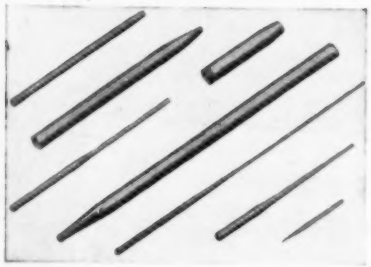

You save time and money when you rely on our high productive capacity and special equipment to turn out precision parts to your order.

Typical are surgical and dental instruments, pen and pencil barrels, soldering iron cases, etc. Medium wall tubing up to 2½" O.D. and solid steel .015" to ⅜" diameter handled.

We are also set up to make such parts as special rollers, shafts, studs, dowel pins, special needles, instrument shafts and pivots, screw driver and ice pick blades, knurled mandrels and spindles, etc.

Send your prints and specifications today for prompt quotation.

THE TORRINGTON COMPANY
Specialty Department
554 Field Street • Torrington, Conn.

Makers of
TORRINGTON NEEDLE BEARINGS

stat, preheater, heater, heater relay insulation and hermetically sealed enclosure. Its output frequency is 400 cycles per second and is constant within 1/10 percent over specified conditions of load, input voltage change, and environmental conditions such as temperature, altitude, salt spray, fungus, sand and dust. This rating, in contrast with the usual practice of specifying frequency deviation "per degree" is for the full range of ambient conditions encountered by aircraft. Special frequency generators, having other rated input voltages (either a-c or d-c), output voltages, output impedances, and frequencies can be built to order.

The standard model is designed primarily as a frequency standard for control of 400 cycle per second electronic inverters, but can be applied to rotary inverters, alternators and vaious signal generators for airborne equipment. It can also be applied as a time standard for counters, clocks, intervalometers and recording oscillographs.

Input is rated at 28.0 (+1.5 —3.0) v d-c, with a 1,200 ma maximum current. Minimum output is 15 v, through the standard load resistance of 1 megohm. Warm-up time is 10 sec, weight is 11 oz, temperature range is —55 to +110 C, and materials and workmanship are according to AN-E-19.

Varo Mfg., Box 638, Garland, Texas

Circle No. 55 on Reader Service Card

Pneumatic Booster Controllers

A pneumatic current controller has been developed to permit the use of standard industrial pneumatic controllers on applications involving electrical components as the final element.

The assemblies can be used in control systems for the ultimate control of temperature, flow, speed, pH, etc. Among their uses would be temperature control of electric furnaces, drying ovens, and electrically heated stills and columns. The controller is also commonly used in conjunction with a small rheostat-potentiometer to regulate the input to electronic speed control devices such as the General Electric Thymotrol and Westinghouse Mototrol. A standard electrical component has been modified for continuous throttling control service, and the cylinder operator includes an integrally mounted positioning device to provide accuracy and sensitivity of positioning to within 1/500 part of its total travel.

Conoflow Corp., 2100 Arch St., Philadelphia 3, Pa.

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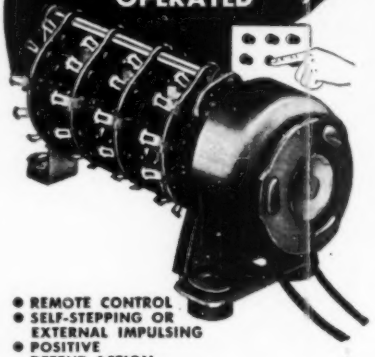
LEDEX

RELAYS and CIRCUIT SELECTORS

ROTARY SOLENOID OPERATED



- REMOTE CONTROL
- SELF-STEPPING OR EXTERNAL IMPULSING
- POSITIVE DETENT ACTION

for REMOTE CONTROL of MULTIPLE COMPLEX CIRCUITS

Many versatile designs of stepping, counting, adding and subtracting, latching, and *circuit selecting* relays are made possible by the combination of the Ledex Rotary Solenoid and wafer type rotary switches. Self-stepped or externally impulsed, the device is immediately adaptable to many remote control applications. A choice of wire sizes permits a wide range of operating voltages and power requirements. Various types of mountings further increase its adaptability. In addition to its positive control of multiple, complex circuits, a reserve of mechanical power is available for the performance of duties other than switching operations.

We supply to quantity users and solicit the opportunity to be of assistance in solving multiple circuit relay problems.

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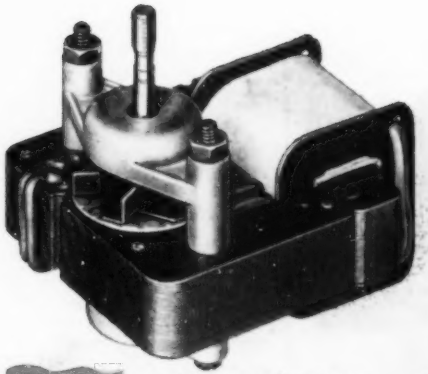


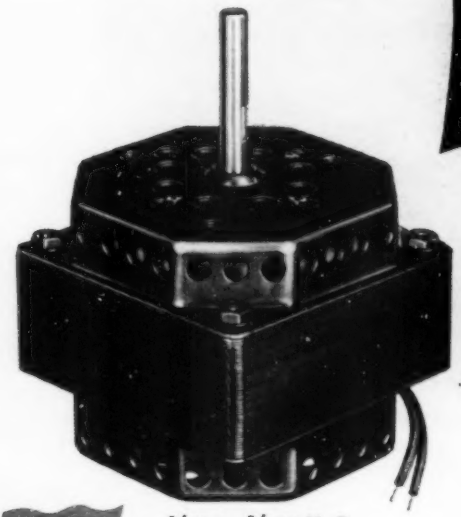
MODEL A . . . 1/125 to 1/200 H. P.

2-pole, shaded pole induction motor. Features include novel bearing construction which insures rigid and permanent alignment of motor shaft . . . oiless bearings.

MODEL C . . . 1/250 to 1/500 H. P.

2-pole, shaded pole induction motor. Features include accurately aligned motor shaft . . . oiless bearings.

SMALL MOTORS WITH A BIG REPUTATION



MODEL D . . . 1/50 to 1/100 H. P.

4-pole, shaded pole induction motor. Features include scientifically designed air intake . . . dual cooling fans . . . self-aligning, oiless sleeve bearings.

It's a time-proved reputation, too . . . for dependability and trouble-free service . . . gained from years of use throughout the radio-phonograph industry.

Whatever your small motor application—action displays, heater or exhaust fans, automatic controls, business machines—you're sure of complete satisfaction when you choose from the GI *Smooth Power* line of small motors.

All three motors available in different sizes to meet power requirements varying from 1/50 to 1/500 H.P. Write *today* for blueprints and quantity price quotations.

The GENERAL INDUSTRIES Co.

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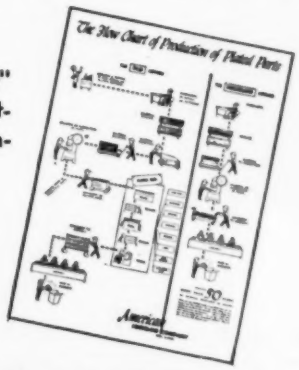
RAW MATERIAL AND FINISHED PRODUCT

Nickeloid *Pre-finished* Metals eliminate many costly operations — cleaning, plating, polishing — and in addition provide lustrous new beauty. You simply *fabricate*, then *assemble!* And those *two steps* take you to lower production costs, *plus* greater product beauty.

Nickeloid Pre-finished Metals are available in sheets or coils, in wide range of gauges, and tempers, plated one or two sides, bright or satin finishes of Chromium, Nickel, Brass or Copper, plated to base metals such as Steel, Zinc, Brass, Copper or Aluminum.

Write for new "Flow-Chart" showing how Nickeloid Pre-finished Metals eliminate costly production operations.



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Electro-plated TO ALL COMMON *Base* METALS
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OLDEST AND LARGEST PRODUCERS OF PRE-PLATED METALS

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OUR READERS SAY

Author's Rebuttal

To the Editor:

In his comments on page 206 of your June issue, upon my recent article on gear tooth durability (March 1950), Mr. D. W. Dudley raises several points well worth further clarification.

Referring to his original designations, my thoughts on the points are as follows:

1. What is the nature of loading on the gear train?

All data were gathered and analyzed on the basis of a unity load factor corresponding to the average smooth (?) industrial load.

When a rough or cyclic variant load is encountered, a marked reduction in life will be noted unless proper compensation is made.

The preferred method is to analyze an experimental or synthesized load curve and to use the root-mean-cube value in evaluating the gearing through my suggestions on life determination. While not strictly vigorous, this has yielded far better results than the application of any of the commonly used service factors.

2. What margin should be allowed for safety?

Surface failure does not cause the immediate stoppage of a gear set. Consequently, no allowance need be made for safety.

The prediction of gear life should prove especially advantageous in the design of equipment having high dependability requirements, since it would enable the initial choice of an adequate gear design.

Although not stressed in the published article, the life values given correspond to the 10 percent failure life used by the anti-friction bearing industry to permit a ready comparison of life values.

3. What materials should be chosen?

The data used were based entirely upon the use of heat-treated alloy steels with very few carbon steels included. The use of any other material would, of course, render the entire method useless.

With the material restricted to heat-treated steel, very little choice could be found between individual analyses

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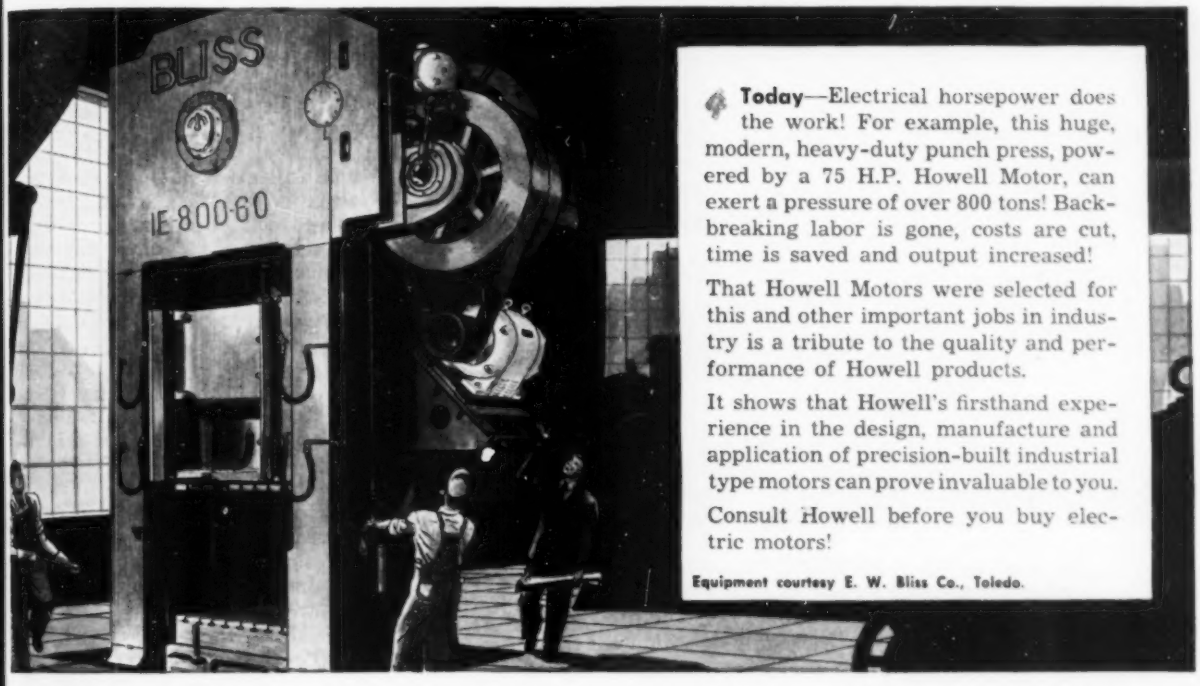
1 **1600**—Man worked hard for his pay in these days. Crude machines like this began to lighten his load by replacing laborious handwork with foot power. But progress was really only beginning.

2 **1875**—By accident, it was discovered that if two direct current dynamos were connected together, one would serve as a generator and drive the other as a motor. Electrical horsepower was here!

3 **1900**—Result: Electrical horsepower replaced foot power! By 1915, Howell "Red Band" Electric Motors arrived. Soon, these rugged, industrial type motors were widely accepted in many industries.

NOW...A PUNCH PRESS WITH A PUNCH!



4 **Today**—Electrical horsepower does the work! For example, this huge, modern, heavy-duty punch press, powered by a 75 H.P. Howell Motor, can exert a pressure of over 800 tons! Back-breaking labor is gone, costs are cut, time is saved and output increased!

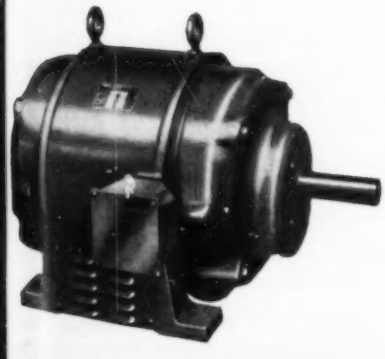
That Howell Motors were selected for this and other important jobs in industry is a tribute to the quality and performance of Howell products.

It shows that Howell's firsthand experience in the design, manufacture and application of precision-built industrial type motors can prove invaluable to you.

Consult Howell before you buy electric motors!

Equipment courtesy E. W. Bliss Co., Toledo.

Free enterprise encourages mass production, supplies more jobs—provides more goods for more people at less cost.



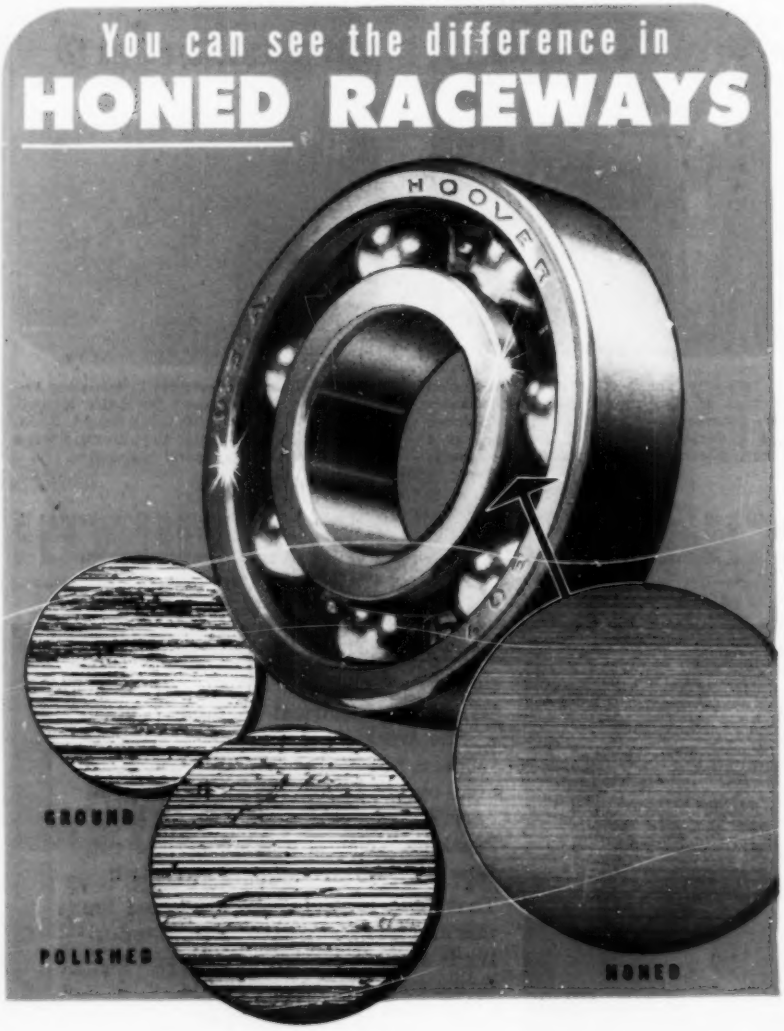
Howell Type F Punch Press Motor

HOWELL MOTORS

HOWELL ELECTRIC MOTORS CO., HOWELL, MICH.

Precision-built Industrial Motors Since 1915

HOWELL Red Band MOTORS



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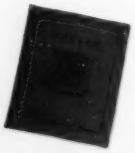
HOOVER BALL BEARINGS

The photographs reproduced above are magnified one hundred times so you can see the difference between ground, polished and honed raceways. The process and the special machines for the honing operation are exclusive, patented, Hoover developments. Honing by Hoover goes far beyond grinding and polishing to achieve a surface that assures amazing quietness plus *90 per cent longer bearing life . . . 30 per cent greater load capacity.* That's why Hoover Ball Bearings are the choice of distinguished American manufacturers of fine machines and equipment.

The Hoover Engineering Manual will be mailed free to engineering and purchasing executives requesting a copy on their business letterhead.



THE ARISTOCRAT OF BEARINGS

HOOVER BALL and BEARING CO.
Ann Arbor, Michigan

provided that they would produce the desired physical properties and accuracy with good manufacturing procedure.

The steel chosen should have adequate hardenability to produce the desired hardness without extreme quenches and/or drawing into a temper brittle range.

In the case of surface hardened gearing, the strength beneath the surface must at all times exceed the applied stress. Generally speaking, the rule of thumb of case thickness (to 50 Rockwell "C") of 0.08 to 0.12 times the tooth thickness at the pitch line has been found satisfactory in preventing sub-surface failure.

Mr. Dudley's comparison of the values of W_d by both Buckingham's method and my modification thereof is questionable, since it presupposes that the total cumulative error in the gear as used to determine C by Buckingham's definition is equal to the individual tooth-to-tooth errors as defined on page 105. While theoretically possible, this condition is highly improbable.

No definite comparison between the value assigned for C for a given gear can be given since C value based on tooth-to-tooth error has been observed to range from 0.1 times that value recommended by Buckingham upwards, approaching unity as a limit.

In practice, ratios between 0.25 and 0.5 have been commonly observed. This means that an error value of 0.001 in. used under my evaluation would probably correspond to a value of 0.002 to 0.004 as defined by Buckingham.

At extremely high speeds, an oil film develops to sufficient thickness to effectively increase the area of gear tooth supporting the load. This results in a drastic reduction in the compressive stress for a given loading. Consequently, either greatly increased life or a considerable increase in load carrying ability is found under this condition.

It is unfortunate that the evaluation of this effect is still so weak that no one can predict its magnitude.

Many other factors of gear design are in much the same position and it is hoped that some day a comprehensive, rational analysis will result.

—Davis M. Wood
Seattle, Washington

Our Sincere Thanks

Ed—Quoted below are exerpts from several of many letters commenting on the special report, "Designing with

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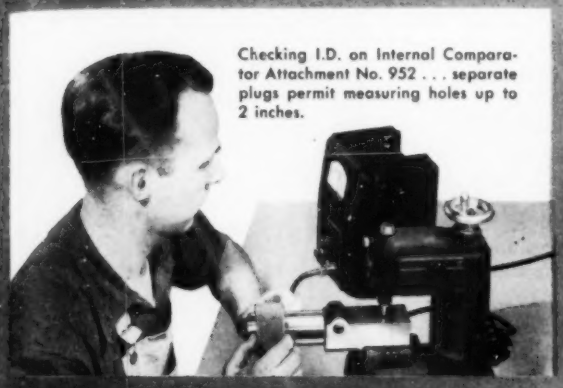
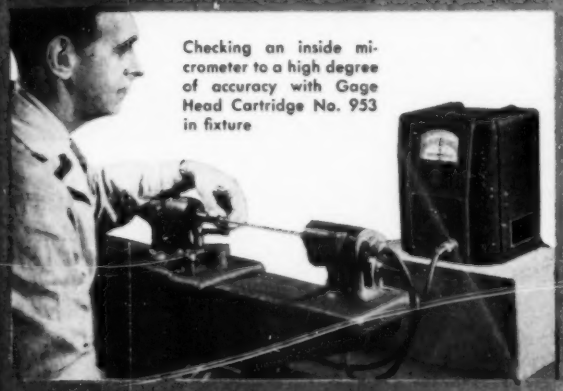
This versatile electronic measuring equipment PAYS ITS WAY IN MANY WAYS



Checking I.D. on Internal Comparator Attachment No. 952 . . . separate plugs permit measuring holes up to 2 inches.

Checking O.D. with External Comparator No. 951, and Amplifier No. 950 mounted on Signal Light Attachment No. 958.



Checking an inside micrometer to a high degree of accuracy with Gage Head Cartridge No. 953 in fixture

Checking gage size with Gage Head Cartridge. Complete freedom from differences in "touch" of different operators.

With this equipment, you can gage a variety of small precision parts on a mass production scale so fast and so accurately that costs drop. No special skill needed. Savings over slower, more tedious methods quickly pay for this equipment.

When used as a part of a Quality Control Program, this Brown & Sharpe Electronic Equipment offers even greater savings. It permits closer, faster coordination of inspection with production . . . making possible a substantial reduction in rejects, wasted time and effort.

This modern gaging equipment is designed and constructed with characteristic Brown & Sharpe precision. For complete information on its unique features, write Brown & Sharpe Mfg. Co., Providence 1, R. I., U. S. A.

UNIQUE FEATURES

- Separate amplifier isolates heat-producing elements.
- True linear response permits accurate setting for entire scale with only 1 master block.
- Single amplifier may be used to serve several testing instruments.
- Normal line voltage fluctuations do not affect readings.
- Simple turn of graduation selector provides accurate readings in any desired increment from .0001" to .00001".
- Simple, self-checking device on external comparator . . . no gage blocks needed.

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Flexible ALL METAL
COUPLINGS
FOR POWER TRANSMISSION
REQUIRE NO MAINTENANCE

Patented Flexible Disc Rings of special steel transmit the power and provide for misalignment and end float.

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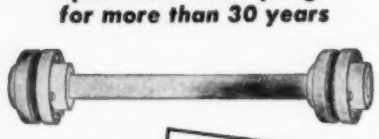
Specialists on Couplings for more than 30 years



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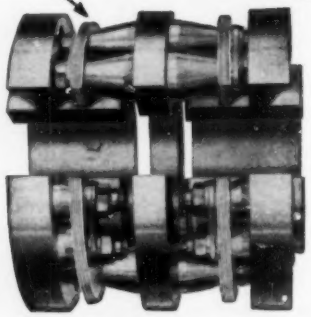

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NO MAINTENANCE PROBLEMS.
ALL PARTS ARE SOLIDLY BOLTED TOGETHER.

Write for the latest reprint of our Engineering Catalog.

THOMAS FLEXIBLE COUPLING CO.
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Dollars", as published in the June issue of PRODCT ENGINEERING.

"Our . . . division which manufactures welding electrodes and gas welding rods has requested that we find out whether it would be possible to get permission to use all or part of the article titled "Welding is Cheaper" on pages 134 and 135 of the June issue.

"What they have in mind is the use of part or all of the article in a single sheet which they send out to distributors and their salesmen. . . ."

"I read with great interest your special report to industry, pages 81 to 144, entitled "Designing with Dollars" in the June issue.

"I am sure this material would be of great value to our draftsmen and engineers and therefore was wondering if it would be possible to obtain a dozen reprints."

"I have just looked through the June issue of PRODUCT ENGINEERING and like very much your editorial and the special report to industry "Designing with Dollars."

". . . our company is greatly interested in product improvement and cost reducing, and I would appreciate very much if you could send two or three tear sheets of the editorial and the report. I would like to circulate them among some interested persons."

A Familiar Chord

To the Editor:

The "Tip Sheet" on page 137 of your June issue "Tubing for Hydraulic Circuits" looks very familiar. Much of the material is obviously from The Parker Appliance Company's "Tube Fitter's Manual"; and the balance of it is practically the table of "Do's and Don't's of Hydraulic Tubing" which I worked up at PRODUCT ENGINEERING's request on behalf of Parker and with the cooperation of Bob Davies, Chief Engineer of Parker Appliance Company. You did not give the Parker Appliance Company a credit line. Finally, item No. 4 (g) of the tip sheet reads, ". . . it is recommended that tube lines be as short as possible. . . ." This should have read, "Distances between tube supports should be not more than 5 feet."

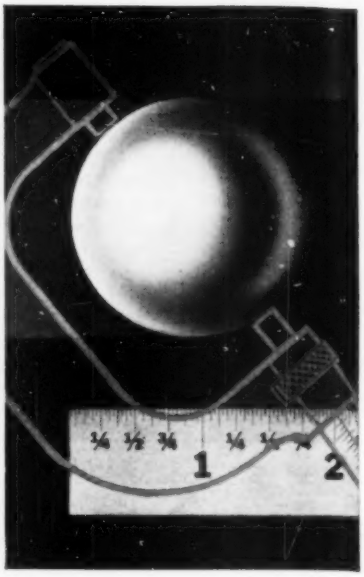
—R. T. MCNAMARA
Hill and Knowlton

Ed.—Mr. McNamara is correct in everything he stated. PRODUCT ENGINEERING is greatly indebted to Bob Davies, Parker Chief Engineer and Bob McNamara of Hill and Knowlton

(Continued on page 222)

ONLY A BALL

has { one dimension / one surface



but oh—how important

Important not only in precision ball bearings, but also in the lot of other applications where Strom metal balls have been doing the job better. Strom has been in on a great many ball-application problems, and knows how important these two factors are for the best results.

Strom has been making precision metal balls for over 25 years for all industry and can be a big help to you in selecting the right ball for any of your requirements. In size and spherical accuracy, perfection of surface, uniformity, and dependable physical quality, there's not a better ball made.

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Investigate the advantages of superior bearing performance available through our exclusive method of sintering copper lead alloys.

In this process each tiny particle of powder is a true alloy of the desired proportions of copper and lead. In the finished bearing this process has created a known and controlled grain structure, of exceptional purity, with excellent lead distribution and with no segregation. Its ductility and strength are superior to cast materials of comparable analyses.

A request on your letterhead will bring a copy of our illustrated technical bulletin describing the development of this improved alloy made possible through powder metallurgy. We believe you will find it a worthwhile addition to your reference file.

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Over 50 Years of Continuous Bearing Experience

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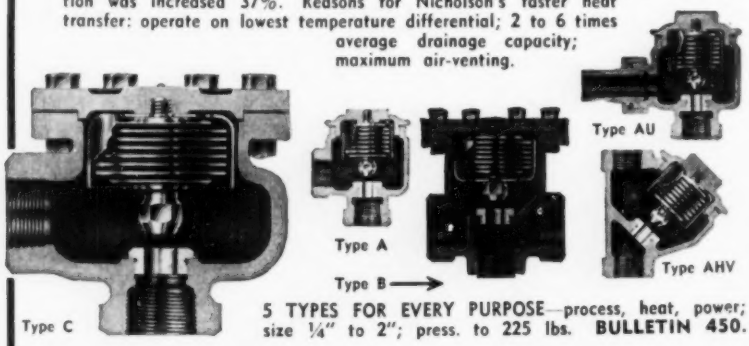
A leading drug manufacturer found that a recent Nicholson thermostatic steam trap installation, replacing a mechanical type, so cut kettle time that their production was increased 37%. Reasons for Nicholson's faster heat transfer: operate on lowest temperature differential; 2 to 6 times average drainage capacity; maximum air-venting.



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HIGH-PRESSURE FLOATS—Stainless, monel, steel or plated steel. Welded. In all sizes and shapes; for operating mechanisms and as tanks or vessels. 2-day delivery. BULLETIN 348.

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Our new molding technique permits us to state unequivocally that:

1. Nylon parts cost less (often considerably less) than metallic components they replace in the majority of cases.
2. Die costs are not expensive.
3. Small runs are possible at a reasonable price.
4. Our prices on large runs are definitely competitive.

We offer the assistance of our research engineers to help solve your problems in connection with Nylon Molding.

If you are not now using Nylon, our product study service is available. Possibly we can show you how advantageous Nylon Molding can be with our methods and equipment.

SCHAEFER-HAUSER MOLDING CORP.

and the wealth of data in the Parker Appliance Company's book, "Tube Fitter's Manual", for the tip sheet on "Tubing for Hydraulic Circuits." The usual credit line was inadvertently omitted in the last minute rush.

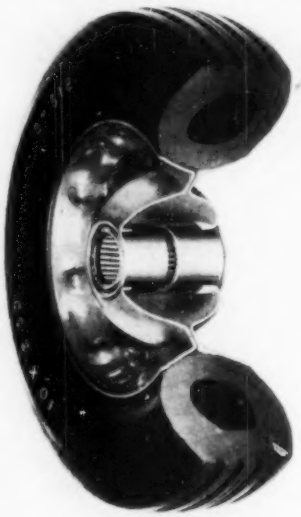
The Needle Is In Order

To the Editor:

In the April issue of PRODUCT ENGINEERING, there is an interesting article on the subject of "Semi-Pneumatic Wheels". On page 121, the wheel labeled "Tapered Roller Bearings" shows two bearings of our manufacture. Actually, these units are needle roller bearings, and we feel some correction regarding proper nomenclature is in order.

—W. S. ST. ONGE
The Torrington Co.

Ed—We are glad Mr. St. Onge has pointed out this error. The original illustration reprinted below demonstrates his point without need for additional comment.



Heat Treatment Is Critical

To the Editor:

We would appreciate receiving more complete information on the hardenability process with chromium-carbon powder for low-carbon steels as mentioned briefly in the Highlights section of PRODUCT ENGINEERING. To which of the low-carbon steels does this process apply and will there be a heat treat variation according to carbon content? —W. A. WITHAM
Lynwood, California

Ed.—This process, called Hard-N-

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An example of the HIGH FATIGUE LIFE you can get in SANDVIK SPRING STEEL

Compressor valves are one of the uses of Sandvik steel demanding extra high fatigue life. In the performance of their extremely vital job in the heart of a compressor, these valves vibrate many, many times per second. The fact that they are required to give years of trouble-free service is evidence of the fatigue resistance you'll find in Sandvik steels.

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This high fatigue life available in Sandvik steels is a result of the basic purity of raw materials used and close controls maintained throughout all processing.

Sandvik strip steels are available:

- **In straight carbon and alloy grades**
- **In special analyses for specific applications**
- **Annealed, unannealed or hardened and tempered**
- ***In thicknesses from .001"***
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- **Unpolished or polished bright, yellow or blue**
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Phone, write or wire your nearest Sandvik office for further information or technical help.

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Allen technical development (available through Allen distributors or direct from the factory) leads the field. We work constantly with engineers of leading manufacturers toward the solution of problems involving fastenings and we invite your inquiry.

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Deep, can be applied to all types of low-carbon steel. Grades of very low carbon content can be heated to 1,650 F without danger of burning, while those of higher carbon content are best limited to about 1,550 F. In the latter case, an oil quench is recommended. The chrome-rich powder is marketed by Miracoloy Corporation, 50 Broadway, New York 4, N. Y.

Now It's Complete

To the Editor:

Somehow it seems rather impossible that the discussion of gearing which appeared on pages 90 and 91 of your June issue could have accidentally omitted reference to the double enveloping gearing in the descriptions carried under worm gears of the different types, specifically excluding Cone-Drive gearing.

We also notice that no mention was made of tungsten carbides among the nonferrous powder metals and I am at a loss to understand why.

—A. F. D.
Detroit, Michigan

Ed.—Obviously, in the paragraph relating to worm gears on page 91, the article should have stated, "There are several types of worm gears, the most common being the type in which the gear partially envelops the worm. In the Cone-Drive gearing, the gear partially envelopes the hourglass-shaped worm, this double enveloping design giving the gearing considerable added load-carrying capacity and making possible extremely short center drives."

Also on page 104, under "Nonferrous" metal powders, the tungsten carbide family of such powders merits more than passing mention particularly in view of their increasing use for wearing resisting parts as well as cutting tools.

One More "B" Line

To the Editor:

The Reference Book Sheet that appeared on page 157 of the June PRODUCT ENGINEERING is complete except for one detail. The resistance of a fully open gate valve is not indicated on the "B" scale. It is equal to that for a sudden contraction, $d/D = 3/4$, as shown at the lower end of the scale.

—E. P. DECRANE
Crane Company

Ed—We regret this omission and hope our readers will note this on file copies of this reference sheet.

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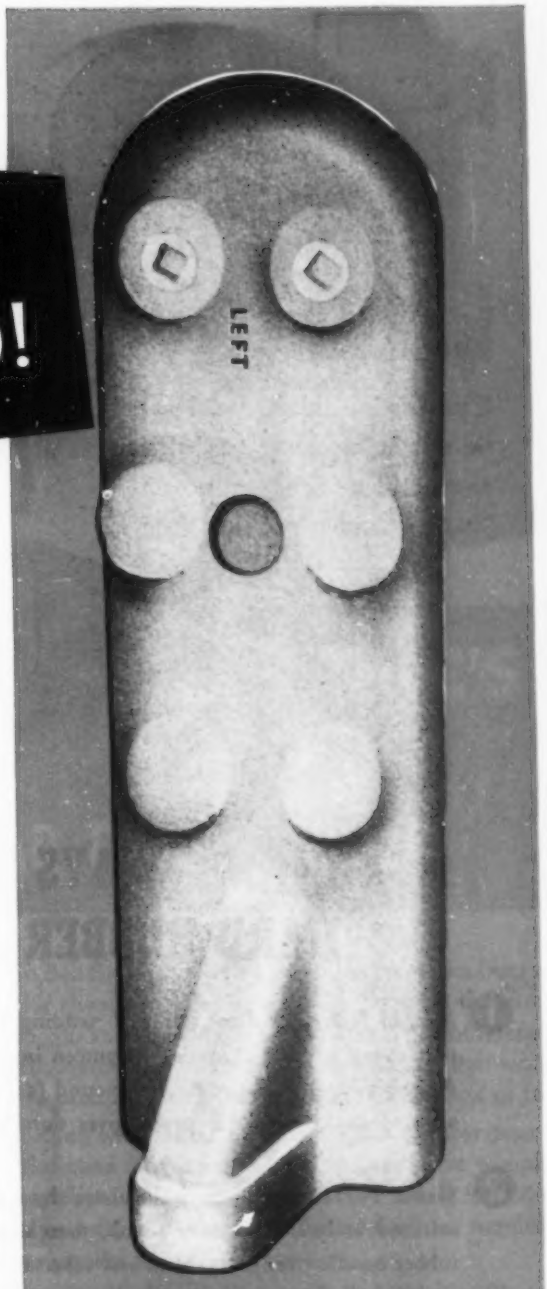
THIS CASTING SAVED $12,000.00!



Why an Electrode Holder, redesigned by National Bearing Division, lasted over 73 times longer

A large Midwest plant was getting 5 days service from furnace electrode holders . . . before a trial order was placed with National Bearing Division. This plant saved dollars—*right from the start.* A special copper alloy permitted sound, dense castings at no sacrifice in electrical conductivity. Result? *30 days service—instead of 5—before replacement.*

On the next order, National Bearing Division engineers submitted a new design for the clamp—for greater strength and resistance to cracking. The redesigned electrode holders are still in use—after a year of trouble-free operation! They've saved $12,000.00 on replacement costs and furnace maintenance.

Better, longer-lasting non-ferrous parts can save money in your plant or product . . .

National Bearing Division has complete facilities for finding practical, economical solutions to non-ferrous bearing and casting problems. Investigate these complete facilities—whether your problem requires designed engineering service, or production of non-ferrous parts to your own specifications.

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Find out how complete facilities for Research, Engineering and Production can reduce costs of non-ferrous parts in your plant or product. Write for a free copy of this catalog.

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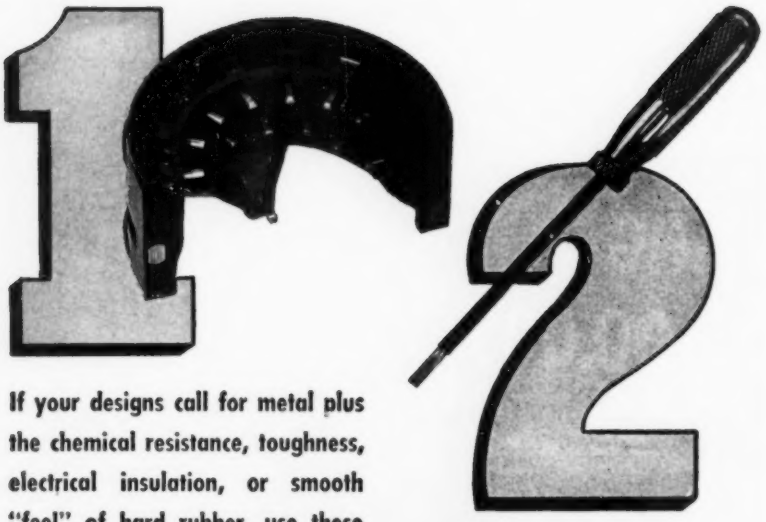
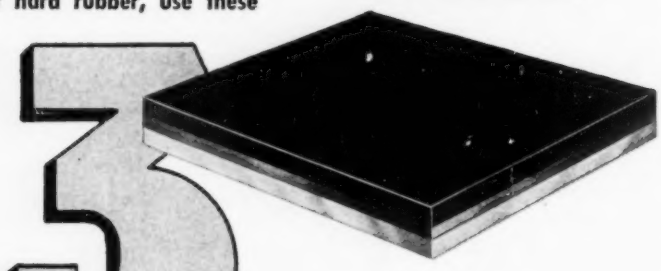


If your designs call for metal plus the chemical resistance, toughness, electrical insulation, or smooth "feel" of hard rubber, use these



3 WAYS TO COMBINE HARD RUBBER AND METAL

1. *Molded Inserts:* Follow standard molding design practice. Magneto part above is good example. Seventeen inserts are molded into high dielectric Magnon Super Ace compound (dielectric strength 600 v/mil at 60 cycles). This Ace grade gives durable strength up to 300°F.

2. *Shrink Fits:* Screwdriver is insulated by slipping hard rubber tube softened in hot water over shank, then slipping heat-softened hard rubber handle over both shank and tube. Rubber cools, shrinks, gives positive-grip, insulated tool as no other material can do. An idea for you?

3. *Vulcanized Cement Bond:* Two-layer process: Hard rubber outer layer for best resistance to chemicals and aging; live, soft-rubber inner layer provides resilience. These Ace linings also can be all-hard or all-soft rubber, natural or synthetic.

Ask for ACE Handbook, a gold mine of helpful information

ACE Since 1851 AMERICAN HARD RUBBER COMPANY
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NEW BOOKS

Evaluation of Effects Of Torsional Vibration

Compiled by the Torsional Vibration Committee of the War Engineering Board of the Society of Automotive Engineers. 578 *pages,* 8½ x 11 *in. Published by the SAE,* 29 *W.* 39*th St., New York* 18, *N. Y. SAE members,* $5; *Non-members,* $10.

Representing the combined efforts of many well-known and highly competent engineers, this study of torsional vibration problems in internal combustion engines was brought about by problems which confronted the Navy Department. Such a comprehensive study was found necessary to better understand and to limit the serious stresses in naval diesel installations where weight and space are important factors.

The book consists of individual contributions from men in industry and is divided into three categories: (1) Various techniques used in measuring torsional vibration; (2) analytical methods for calculating torsional stresses; and (3) significance of stresses and devices used to dampen vibration. Since most of the discussion centers around the methods and practices of the internal combustion industry, it represents the most recent and practical approach to this problem.

Engineering for Production

WALTER ERNST. *Spiral-bound,* 200 *pages,* 8½ x 11 *in. Published by the Research Press, Inc.,* 139 *N. Perry St., Dayton, Ohio.* $10.

There are often conflicts between what engineers want and what the production shop produces. In many cases, the difficulty arises as a result of improperly stipulating design data on drawings. This type of problem wastes not only time, but material as well. It can be extremely costly, especially in purchasing and subcontracting work. Complete and consistent drafting procedures can prevent the occurance of this type of problem.

This book has been prepared as a guide or tool to insure that engineering designs are properly manufactured. This is the result when shop people and production men have no trouble working with drawings.

For the designer and draftsman,

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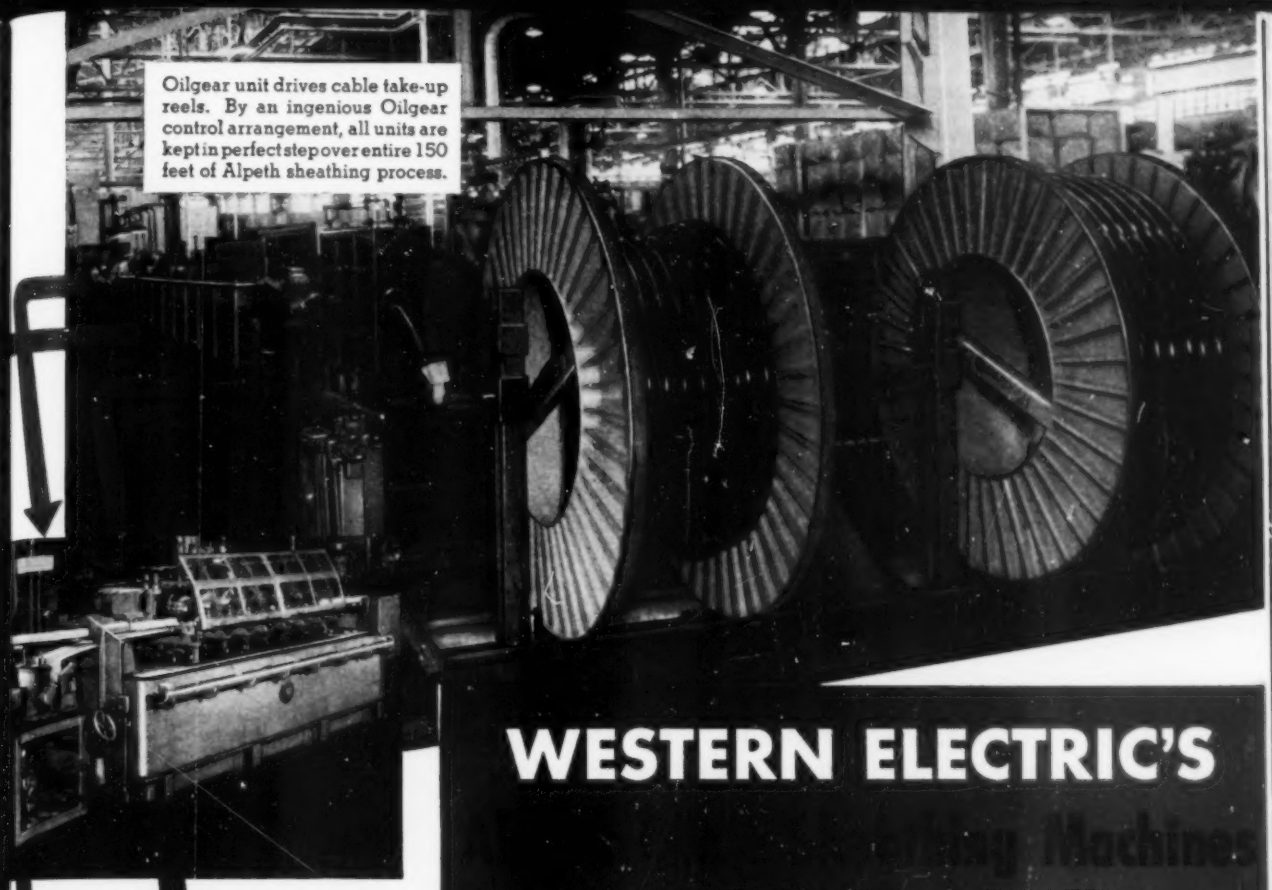
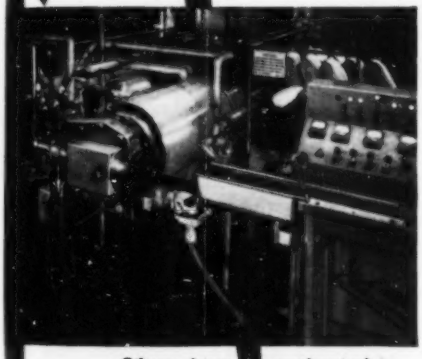


Oilgear unit drives cable take-up reels. By an ingenious Oilgear control arrangement, all units are kept in perfect step over entire 150 feet of Alpeth sheathing process.

Oilgear fluid p[illegible]r drives Yoder roll forming machine which [illegible]aps corrugated aluminum sheathing arou[illegible]able.



Oilgear drives [illegible] synchronized to speed of Royle e[illegible]der which covers cable with melt[illegible]lyethylene.

[illegible]ctor capstan powered by Oilgear, which pulls cable [illegible]rough extruder, long cooling trough and inspection [illegible]ion.

WESTERN ELECTRIC'S

OILGEAR FLUID POWER drives

• Right after the war, the Bell System faced a tremendous problem because of the shortage of lead for sheathing telephone cable. To solve that problem, a newly designed sheath known as Alpeth—aluminum polyethylene—was adopted. And the engineers had to design a machine fast that would apply this sheath fast—to produce this new cable by the millions of feet.

Oilgear engineers, working with *standard* Oilgear Fluid Power pumps, motors and valves, met the need for drives that would handle the complex Alpeth coating machines, dependably, without trouble . . . and for drives that could be immediately, easily and economically installed . . . requiring minimum space, sure performers from the start.

The first Alpeth cable sheathing machines installed by the Western Electric Company proved themselves and more units were installed, to make a total of three at the Kearny, N. J., Works and three at Hawthorne Works in Chicago. The Oilgear drives have performed well, without trouble, with minimum maintenance, often at high speed for 24 hours a day.

A company like Western Electric does not buy by guesswork. Neither should you. THE OILGEAR COMPANY, 1571 W. Pierce Street, Milwaukee 4, Wisconsin.

Oilgear

PIONEERS IN FLUID POWER

PUMPS, MOTORS, VALVES, CYLINDERS

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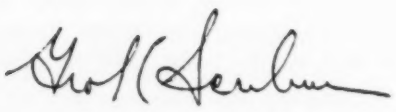
MOLDS FOR PLASTICS!

IT SOUNDS FOOLISH to say that molds are what you make them, but they are, and can be a definite headache for that reason. When a customer buys the cheapest mold that is offered to him, he may easily be heading for trouble. That same purchasing agent would throw up his hands in horror if some one proposed that he buy the cheapest available truck to tote his goods around. He knows well that there are other things to trucks than price. He would say that upkeep and continuity of service were important too, and that he was willing to pay for the insurance of these two items in the original cost.

Because the mold is in some one else's shop, and that some one else is assuming the responsibility for these items, is not enough to justify ignoring the quality of the mold to be purchased. Regardless of the blame being placed elsewhere for a breakdown in production, the purchasing agent will be the one to hear from the front office.

The most talented molder in the world cannot make a bad mold stand up to a heavy schedule. In fact 90% of molding troubles, from accuracy to ultimate delivery, are traceable to inferior molds or mold design. And most of these in turn trace back to pressure from some ambitious client who decided he would save money on the molds after the piece price had been settled. Add to that, of course, a weak, hungry molder who yielded in the hope that this time he could get away without trouble. Some time he does.



George K. Scribner, President
BOONTON MOLDING CO., BOONTON, N. J.
Phone: BOonton 8-2020

there is a considerable amount of handy reference material. It is particularly informative in the discussion on metals and their heat-treatment. The material presented is patterned on data from the American Standard Association, the American Welding Society, and the Joint Industry Conference on Hydraulics. Included are sections on piping, packings, gears and bearings.

Kent's Handbook

Power

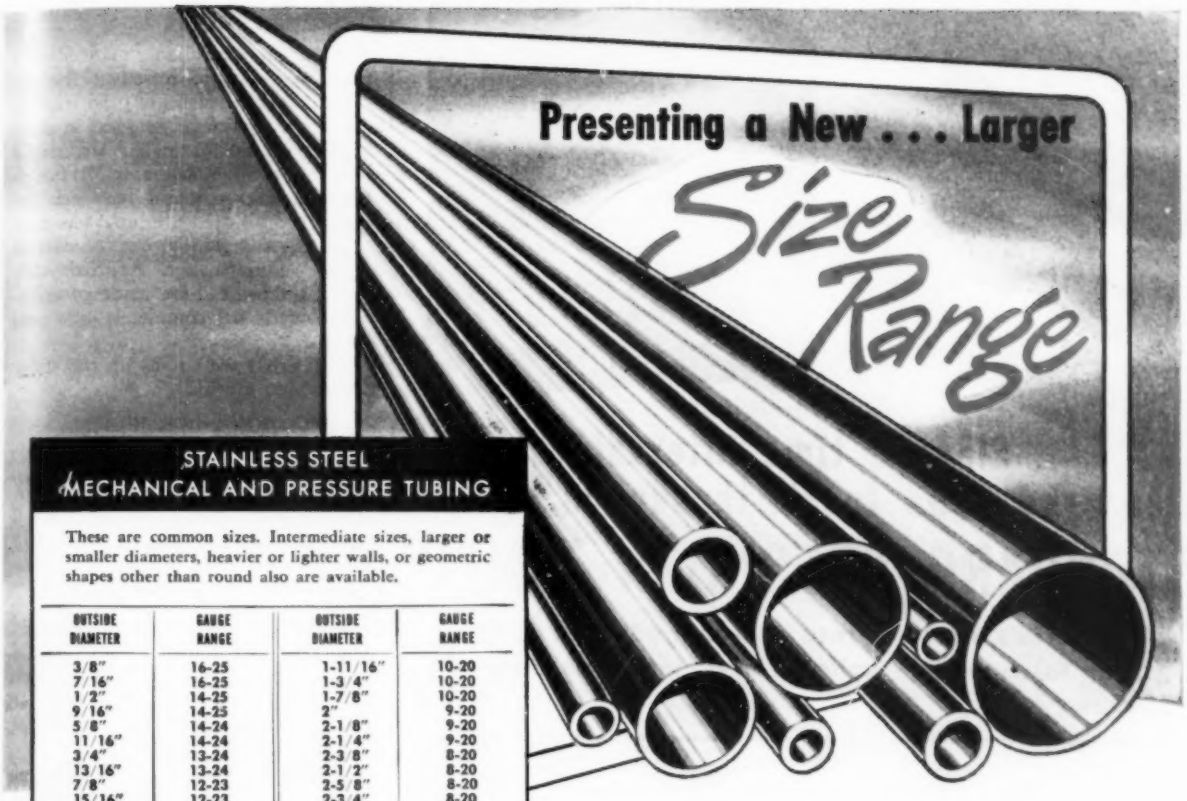
Edited by J. KENNETH SALISBURY, *Division Engineer, General Engineering and Consulting Laboratory, General Electric Co. Twelfth Edition, 1455 pages, 5½ x 8⅜ in. Published by John Wiley & Sons, 440 Fourth Ave., New York 16, N. Y. $8.50.*

What is true of the Design and Production volume of this handbook is also true of the Power volume: it is a comprehensive reference that will become an invaluable part of any engineer's bookshelf. Inclusion of data on such subjects as atomic power, combustion gas turbines, supersonics, marine engineering, jet propulsion, axial piston engines, heat pumps, panel heating, diesel and electric locomotives, axial flow compressors, instrumentation and on other recent developments really makes previous editions of the Power volume obsolete.

Space devoted to steam engines, gas producers, and power test codes in the past editions has been drastically cut in this one. Mathematical tables have generally been retained and amplified by data on geometry, calculus, differentiation formulas, and tables of integrals.

Sections in this, the twelfth edition, are as follows: (1) Air; (2) Combustion and Fuels: (3) Heat and Heat Exchange; (4) Steam, Water, and Ice; (5) Hydrodynamics, Hydraulics, and Pumps; (6) Piping; (7) Steam Generating Units; (8) Steam Turbines and Engines; (9) Condensers and Cooling Equipment; (10) Combustion Gas Turbines; (11) Refrigeration and Ice Making; (12) Heating, Ventilating, and Air Conditioning; (13) Internal Combustion Engines; (14) Land Transportation; (15) Air and Marine Transportation; (16) Electric Power; (17) Atomic Power; (18) Instrumentation; (19) Power Test Codes; and (20) Mathematical Tables.

Quoting from the preface of the book: "Advances made in mechanical engineering during the twelve years since publication of the eleventh edi-



STAINLESS STEEL MECHANICAL AND PRESSURE TUBING

These are common sizes. Intermediate sizes, larger or smaller diameters, heavier or lighter walls, or geometric shapes other than round also are available.

OUTSIDE DIAMETER	GAUGE RANGE	OUTSIDE DIAMETER	GAUGE RANGE
3/8"	16-25	1-11/16"	10-20
7/16"	16-25	1-3/4"	10-20
1/2"	14-25	1-7/8"	10-20
9/16"	14-25	2"	9-20
5/8"	14-24	2-1/8"	9-20
11/16"	14-24	2-1/4"	9-20
3/4"	13-24	2-3/8"	8-20
13/16"	13-24	2-1/2"	8-20
7/8"	12-23	2-5/8"	8-20
15/16"	12-23	2-3/4"	8-20
1"	11-23	2-7/8"	8-20
1-1/16"	11-22	3"	8-20
1-1/8"	11-22	3-1/4"	8-16
1-3/16"	11-22	3-1/2"	8-16
1-1/4"	11-22	3-3/4"	8-16
1-5/16"	11-20	4"	8-16
1-3/8"	11-20	4-1/4"	8-16
1-7/16"	11-20	4-1/2"	8-16
1-1/2"	10-20	4-3/4"	8-16
1-9/16"	10-20	5"	8-16
1-5/8"	10-20		

The above sizes are available in types 304, 309S, 310, 316, 317, 321, 347 and 430. Type 302 is available in a limited range of sizes for ornamental use.

STAINLESS STEEL PIPE

NOMINAL PIPE SIZE	OUTSIDE DIAMETER	SCHEDULE 40S STANDARD PIPE WALL THICKNESS	SCHEDULE 10S LIGHT WALL PIPE WALL THICKNESS
1/8"	.405	.068	.049
1/4"	.540	.088	.065
3/8"	.675	.091	.065
1/2"	.840	.109	.083
3/4"	1.050	.113	.083
1"	1.315	.133	.109
1-1/4"	1.660	.140	.109
1-1/2"	1.900	.145	.109
2"	2.375	.154	.109
2-1/2"	2.875	—	.120
3"	3.500	—	.120
3-1/2"	4.000	—	.120
4"	4.500	—	.120

The above Pipe Sizes are available in the Chrome-Nickel analyses of Republic ENDURO Stainless Steel.

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Stainless Steel Tubing and Pipe

● NOW . . . up to and including 5" O.D. . . . ELECTRUNITE Tubing made of time-tested Republic ENDURO* Stainless Steel.

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Republic—world's largest producer of stainless and alloy steels—offers you prompt metallurgical assistance with any problem involving the use of stainless steel tubing or pipe. Write, wire or phone *today* for further information.

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ELECTRUNITE
STAINLESS STEEL TUBING AND PIPE

Elapsed time intervals can be measured down to 1/1000th second with a

STANDARD PRECISION TIMER

The world's most accurate (and rugged) time measuring instrument. Available in 8 models as shown below:

MODEL	READS	TOTALIZES	ACCURACY
S-100	1/5 sec.	6000 sec.	± .1 sec.
S-60	1/5 sec.	60 min.	± .1 sec.
SM-60	1/100 min.	60 min.	± .002 min.
S-10	1/10 sec.	1000 sec.	± .02 sec.
S-6	1/1000 min.	10 min.	± .0002 min.
S-1	1/100 sec.	60 sec.	± .01 sec.
MST	1/1000 sec.	.360 sec.	± .001 sec.
MST-500	1/1000 sec.	30 sec.	± .002 sec.

Manually and Electrically Operated Portable Cases or Panel Mounting optional changes to conform to customers' requirements.

Special Timing Problem. **Our engineers are ready to consult with you, and design timers to meet special timing problems.**

OTHER PRODUCTS

Standard Chrono-Tachometers, Custom-built Laboratory Test and Distribution Panels, School Time Program Systems, Hospital Doctor Paging and Nurse Call Systems.

WRITE FOR BULLETIN NO. 153

STANDARD

THE STANDARD ELECTRIC TIME CO.

95 LOGAN STREET **SPRINGFIELD, MASSACHUSETTS**

tion of the Handbook made this a propitious time to introduce the expanded twelfth edition. So rapidly has technology raced forward in these years that the book has necessarily been enlarged considerably. Although much obsolete material has been deleted, the size of the volume has been increased about 20 percent by addition of new subject matter. Approximately eighty authorities have made contributions, nearly all completely new and original."

That fairly well sums up the status of this new edition.

The Nature of Physical Reality

HENRY MARGENAU. *479 pages, 6 x 9 in. Published by the McGraw-Hill Book Co., Inc. 330 W. 42nd St., New York 18, N. Y.* $6.50.

Implications of the remarkably rapid progress being made in the physical sciences are described in this book by Henry Margenau, Professor of Natural Philosophy and Physics at Yale.

Students of physics and theory of knowledge should be grateful for the clarity and thoroughness of this competent analysis of the basic ideas which are the tools of research. Instead of imposing fixed rules of method or value-judgments on scientific thinking, Mr. Margenau emphasizes the basic lesson of modern physics: Ideas that are valid for one type of experience should not be extended to all or new kinds.

Electrical Machines, Direct And Alternating Current

CHARLES S. SISKIND, *assistant professor of electrical engineering, Purdue University. 521 pages, 6 x 9 in. Published by the McGraw-Hill Book Co., Inc., 330 W. 42nd St., New York 18, N. Y.* $4.75.

The constructional and operating similarities, as well as their differences, of direct and alternating current machines are treated with special emphasis in this book, especially as they apply to modern electrical machinery practice. Similar or dissimilar characteristics are compared, and some constructions are shown that are derived from, or modifications of, the fundamental types.

The material presented presupposes a knowledge of elementary direct and alternating current circuit theory and practice. The mathematical treatment is limited to simple arithmetic, alge-

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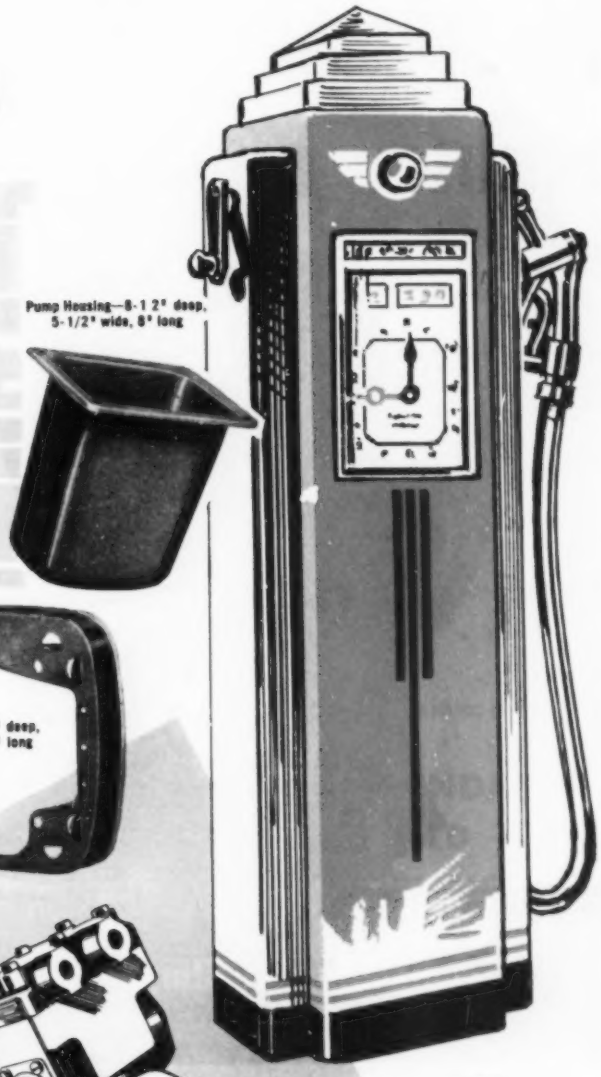
IMPROVED BY T and W DEEP DRAWN STAMPINGS

● Strength, shock resistance, reduced weight and lower cost were all achieved by these T & W Steel Stampings. Perhaps you, too, have similar problems that can be solved by redesigning. This Transue engineering service is available to you for investigating the possibilities of redesign.

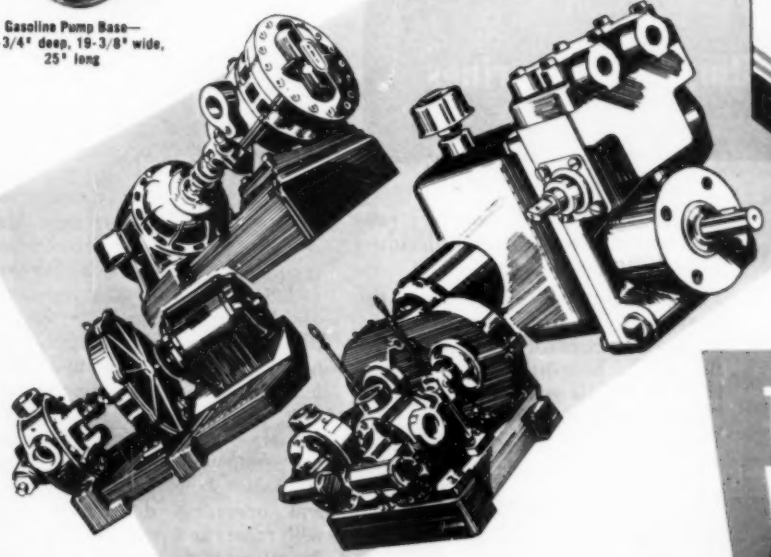
NO. 4 OF A SERIES



Pump Housing—8-1 2° deep, 5-1/2° wide, 8° long

Pump Base—3-1/4° deep, 12-1/4° wide, 22° long

Gasoline Pump Base—13-3/4° deep, 19-3/8° wide, 25° long



For those manufacturers having their own press equipment, Transue will gladly quote on die requirements.

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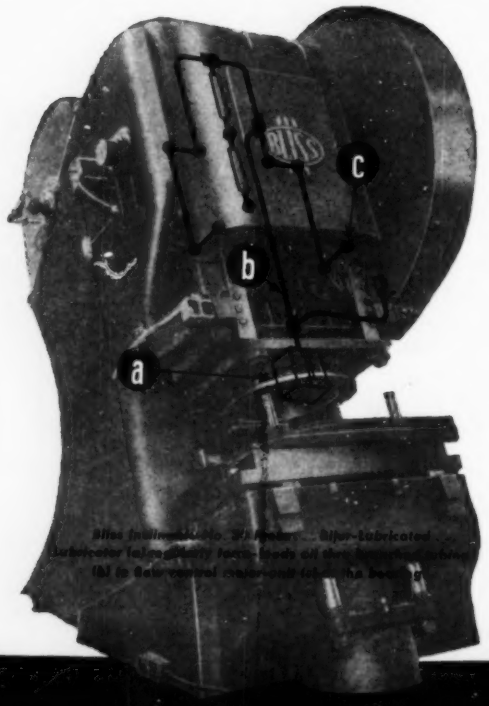
DESIGNERS AND MAKERS OF DEEP DRAWN STAMPINGS

LEADING MANUFACTURERS STANDARDIZE ON THE BIJUR SYSTEM

reduce bearing wear



by controlling oil film at the bearings

Oil film at the bearings keeps the metallic surfaces apart, reducing wear to a minimum. This oil film must be maintained *constantly* to be effective.

Lubrication by hit and miss methods can't be depended upon to keep bearings running smoothly. Proper lubrication requires a system which force-feeds the correct amount of oil to all bearings and carefully controls the oil flow at each individual bearing *automatically*.

This is accomplished by Bijur, the system with positive Meter-Unit control of oil flow *at the bearings*. For further details write for "The A B C of Modern Lubrication."

The correct oil film to each individual bearing... *automatically*

387

BIJUR LUBRICATING CORPORATION

LONG ISLAND CITY 1, NEW YORK

bra, and a few trignometric functions. Although in the discussion of the performance of alternating current machines, where phenomena such as armature reaction, armature and leakage reactance, and the revolving field are discussed, some use is made of the vector notation in its elementary form.

Numerous problems are presented and solved. In the main, examples have been chosen from practice with the objective of developing a good sense of judgment with regard to machine sizes, operating characteristics, magnitudes of electrical quantities, and the proper selection, installation, and operation of machinery.

Motor Oils And Engine Lubrication

CARL W. GEORGI, *Technical Director, Research Laboratory, Quaker State Oil Refining Co.* 514 *pages,* 6 x 9 *in. Published by Reinhold Publishing Corp., 330 W. 42 St., New York* 18, *N. Y.* $8.50.

The practical problems of engine usage, maintenance and lubrication, as well as causes and remedies affecting operating troubles, are described in this comprehensive book. In it are assembled for ready reference detailed information on the properties and service behavior of motor oils.

It covers at length the methods of testing and evaluating performance characteristics of motor oils. Particular emphasis is laid upon viscosity index and its importance in all phases of motor lubrication. A feature of the book is the inclusion of detailed specifications covering various types of engine tests, properties of lubricants (ash, carbon residue, pour point, color, corrosion, flash point) and all types of performance ratings. Also included are chapters devoted to the refining and manufacture of motor oils and description of their chemical and physical properties. Later sections of the book deal with the application of motor oils and the relation of oils and lubrication to engine design, operation, maintenance, breakdowns and failures. The subject matter is arranged to permit ready reference to many types and varieties of engine failures and operating difficulties, together with causes and remedies. Valuable information is given to aid in diagnosing engine ailments, and in adopting corrective steps necessary to prevent their recurrence.

The text has purposely been written in practical terms and avoids as much

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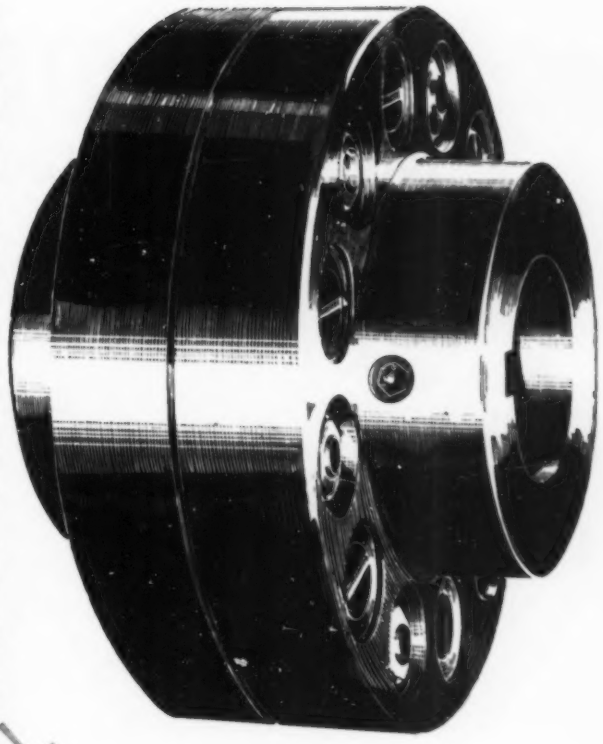

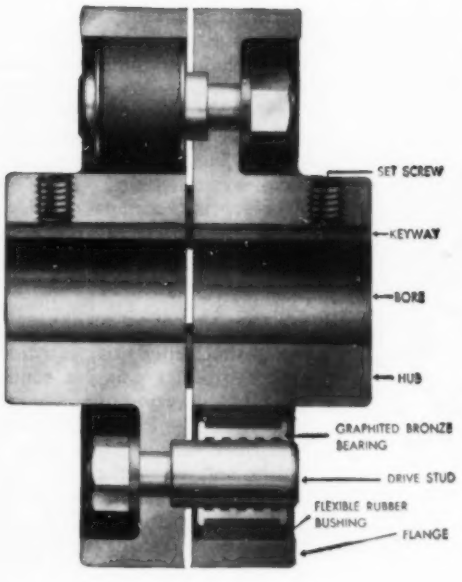

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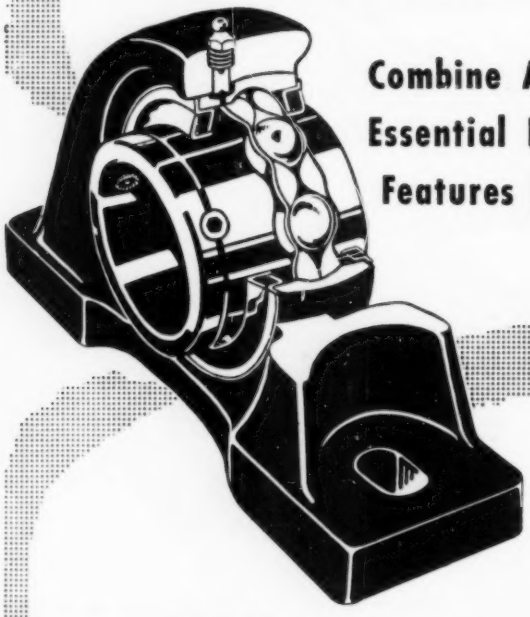

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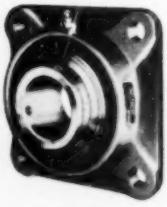

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as possible highly involved technical considerations. Illustrations are used freely, especially those relating to engine wear. The book has been written primarily as a reference work.

Numerical Methods of Analysis in Engineering

Compilation of papers given at a symposium held at Illinois Institute of Technology. Arranged and edited by L. E. GRINTER. 207 *pages,* 6¼ x 9½ *in. Published by the Macmillan Co., 20 Fifth Ave., New York, N. Y.* $5.80.

This is a collection of ten papers on the use of numerical methods to solve engineering problems. The first group entitled "Numerical Methods Based upon Physical Concepts" consists of two papers. The first is a reprint of the famous paper by Hardy Cross on moment distribution. The second, written by L. E. Grinter, is entitled "Statistical State of Stress Studied by Grid Analysis". This gives a method of calculating approximate stress distributions. In justifying this method, the author emphasizes the errors in the assumptions of the theory of elasticity rather than its relationship to his technique. This paper contains an appendix by A. J. Pyka applying the technique to the problem of a notched beam.

The second group, "Numerical Solutions of Equations for State of Stress", consists of three papers. The first, by F. S. Shaw, discusses relaxation techniques for partial differential equations, especially Poissons equation. This paper would be clearer if the techniques were explained in greater detail. The second paper, by R. V. Southwell is entitled "The Quest for Accuracy in Computations Using Finite Differences". He points out that there are two schools of thought on how accuracy should be improved. One school advocates the use of more terms in the formula approximating the partial differential equation. The other advocates using a simple formula but covering the region under consideration with a finer mesh. The last paper in this group is by M. M. Frocht and is entitled "A New Approach to the Numerical Solution of Laplace's Equation". This gives a technique for calculating the solution at any given point of the region under consideration. It also shows how the solution throughout the entire region can be obtained from values at a relatively few points. Combining these techniques is advocated.

The third group, entitled "Applications of Numerical Methods to Heat

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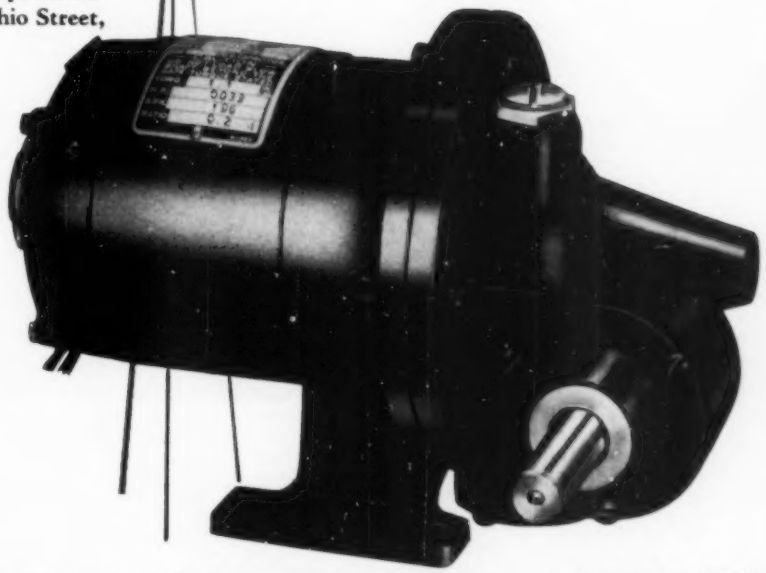

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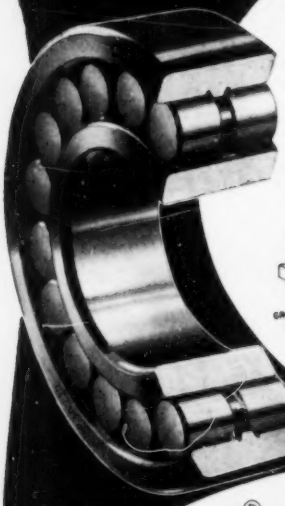

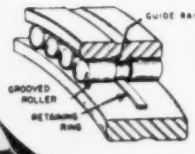

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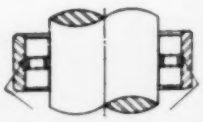

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Transfer," contains two papers. The first, by L. M. K. Boelter and Myron Tribus, gives a technique for calculating the temperature at any instant at any point in a one-dimensional region. The other by G. M. Dusinberre deals with transient and steady flow of heat. While the emphasis is upon problems in which the heat transfer is between different objects, the appendix illustrates a problem worked by J. D. Bottorf and Y. S. Touloukian in which the heat transfer is between different regions of a given object.

The last group entitled "Surveys and Bibliographies of Numerical Methods" contains three papers. The first by Frank Baron is titled "Successive Corrections—a Pattern of Thought". It points out some similarities between problems arising in different fields and methods of solving them. The second paper by N. M. Newmark covers similar ground and also explains a group of techniques for plane problems in the theory of elasticity. The final paper by Thomas J. Higgins is entitled "A Survey of the Approximate Solution of Two-Dimensional Physical Problems by Variational Methods and Finite Difference Methods". Among other techniques, the Rayleigh-Ritz method and the method of Trefftz are mentioned.

Materials Under Low Temperature Conditions

Symposium on the Effect of Low Temperatures on the Properties of Materials. Special technical publication No 78. 68 *pages,* 6 x 9 *in. Published by the American Society for Testing Materials,* 1916 *Race St., Philadelphia,* 3, *Pa.*

This symposium was originally held at a meeting in 1946, and the data was reviewed by the authors in October, 1949. This material covers specifically the fields of both ferrous and non-ferrous metals, plastics and rubber. This material should be of particular interest to those concerned with problems in refrigeration, chemical processes and related fields. Gathered in this booklet are the latest data and experience concerning these materials under conditions from 50 to 100 degrees below zero Fahrenheit. The discussion of organic plastics is given by Messrs. H. K. Mason, T. S. Carswell, and C. H. Adams. Mr. John W. Liska is the author of the paper on elastomers; the properties of non-ferrous metals is discussed by Mr. Norman L. Mochel; and, finally, Mr. Robert D. Stout discusses the properties of weldments at low temperatures.

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